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# A cogas propulsion cycle with peak efficiency at a low power.

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by

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## Abstract

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This thesis is an engineering analysis of a proposed propulsion cycle for a destroyer type ship. It is organized and written in the same manner that an engineer would approach the problem. It starts with the requirements for a destroyer propulsion system and proceeds through a cycle selection process and individual component selection. At each step in the process the feasibility of each component is analyzed. After all the components have been selected, a heat balance is made to ensure that they all fit together. This is followed by a weight, volume, and reliability analysis. Much of the more detailed and voluminous material is included in appendicies to retain continuity in reading the main body.

Thesis Supervisor: A. D. Carmichael
Professor of Power Engineering

## Acknowledgements

Each person who helped in the research and development of this cycle and thesis is recognized at the appropriate spot in the text. But there are several men who gave much more aid than can be noted in this manner.

Mr. William H. VanCott, Marine Consultant to General Electric Company, conceived this cycle and initially spent many hours discussing the thesis with me. He suggested many alternatives at each step in the development of the thesis.

Mr. A. O. White, Manager of Marine Products of the General Electric Gas Turbine Department, spent two full working days with me at the onset getting the gas turbine portion started. In addition, he put me in touch with the proper men to get reliable information on heat recovery, combustion, etc.

Mr. David Gray and Mr. Carl Horlitz of Combustion Engineering spent a great deal of time discussing the boiler portion of the cycle. Additionally, Mr. Horlitz actually sat down and designed the boiler used and made the modifications necessary to use it in this cycle.

Mr. Chester W. Stott, Senior Engineer of The General Electric Steam Propulsion Group, spent many hours with me during the heat balance calculations. Without his knowledge and assistance the heat balance would have been very difficult indeed.

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## Introduction

Traditionally, destroyer type ships are designed for the full power, maximum auxiliary load condition. Actually, these ships spend up to 90% of their underway time at speeds below 20 knots and much of this time is at speeds below 15 knots. A typical destroyer or modern destroyer escort seldom has an electric load exceeding 800KW and almost never exceeding 1000KW. When all the combat equipment is in operation, the electric load is at the rated capacity, but only then.

Using this argument, destroyer type ships should be designed for peak efficiency at a power level corresponding to about 20 knots with an electric load sufficient to cover the normal cruising requirements. In the case of destroyer escorts ships, this cruise power should correspond to the speed most commonly used in convoy or escort duties. Non-classified information shows this power level to be about 4500 SHP for a single screw DE at 15 knots and about 6000 SHP per shaft for the twin screw DD at 20 knots.

Specific fuel rate is not the only factor to consider in naval ship propulsion. Some other important factors are: volume, weight, reliability, maintainability, availability and response time to reach full power from various conditions. Each propulsion system is best in one or more of these areas and no propulsion cycle is best in all. For example, the gas turbine is by far the best when one considers response time and availability. The aircraft gas turbine is by far the best if one is concerned about weight and volume of propelling machinery, but this is offset somewhat if one examines weight and volume of the entire machinery package and cruise fuel load. The diesel engine is by far the most economical

user of fuel, but it is very large and noisy at the power levels necessary for a destroyer.

The combination of all these factors have led the naval engineer to look at combined power plants in order to achieve the optimum for his design. Some of the combinations considered have been:

Gas Turbine and Diesel (CODOG, CODAG)

Gas Turbine and Gas Turbine (COGAG, COGOG)

Gas Turbine and Steam (COGAS)

Proponants of each of these systems have expounded at length on the virtues of each in the literature. But to date no one has studied the combination proposed in this thesis.

This combination consists of a relatively small gas turbine for base (cruise) load using waste heat recovery. There is a relatively large steam boost to the full power requirements. At the cruise mode the boiler is unfired and the steam pressure low. At full power the boiler is fully fired and the steam pressure high.

All the components of this particular combination are available off-the-shelf at this time or are easily designed and manufactured. None of the equipment used in this cycle is beyond the current state of the art. Suggested improvements are made at each step in the process. Some would greatly improve the cycle if they were possible now.

At first glance this cycle looks complicated and confusing. But a deeper analysis shows that it is quite simple and could be made even simpler as the state of the art improves.

## Procedure

The procedure used in the development of this thesis is much like that used by any marine engineer in the development of a concept. Once the basic idea for the cycle was formulated, components had to be found which were suited to it. This process included many inquiries made to individual experts in each of the fields involved. In some cases actual visits to design locations and manufacturing sites were necessary in order to provide a good flow of information both ways. In other cases telephone calls were sufficient.

In the development of a propulsion cycle there are several components which are very difficult to design and require extremely long lead times. In these cases it is mandatory to use existing equipment and to design the rest of the plant around them. This cycle had to be designed around an existing gas turbine and had to use a steam turbine which was only slightly modified from existing design. The boiler design is a modification of an existing design.

After the major components were selected and sized, a determination had to be made on electric power generation and auxiliary equipment. These were sized to fit the cycle and the decision made on the source of power to each.

Once all the components were sized, a heat balance was made. This step required making a few modifications to the cycle before a satisfactory result was obtained. Then a weight, volume and reliability study was made to provide some comparative numbers to use if this plant is to be compared with others.

Alternative suggestions are discussed at each step in the process and future state of the art suggestions are discussed in these steps if appropriate there. If not, they are left for the recommendations and discussion sections.

# Results

## Speed Profile

In a ship propulsion cycle, the first step is to determine the power level of the design points. In order to do this, the speed profile of the ship has to be known or assumed. Figure 1 shows the speed profile which was assumed for this study. It is based on assumption using the following reasoning.

The speeds from 0 to 10 knots are assumed as one block because the power level is so flat at these speeds for a destroyer type ship. There is a peak at the most efficient speed which is typically 14 or 15 knots since this can be assumed to be the speed most used in an independent transit. There is another peak at about 24 to 25 knots as this is currently about the maximum sustained speed using two boilers on a destroyer type ship. The rapid fall-off above 25 knots merely shows that these ships seldom operate in this speed range. The reasoning for this fall-off is that the power requirement increases so rapidly at these speeds that it would be uneconomical to run there unless operational requirements made it unadvoidable.

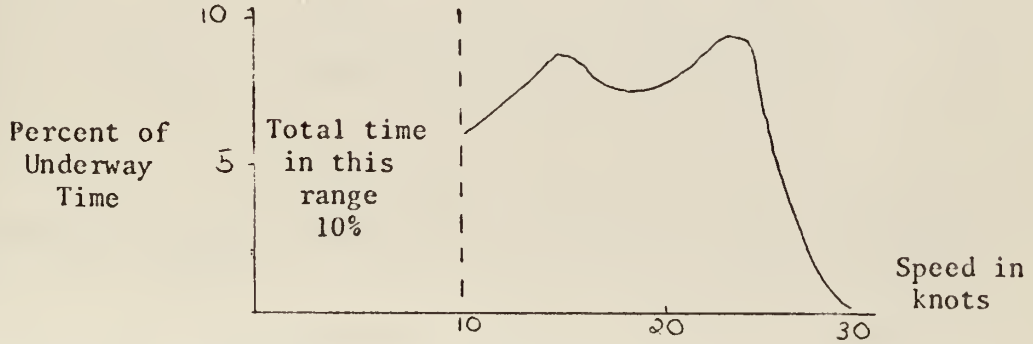
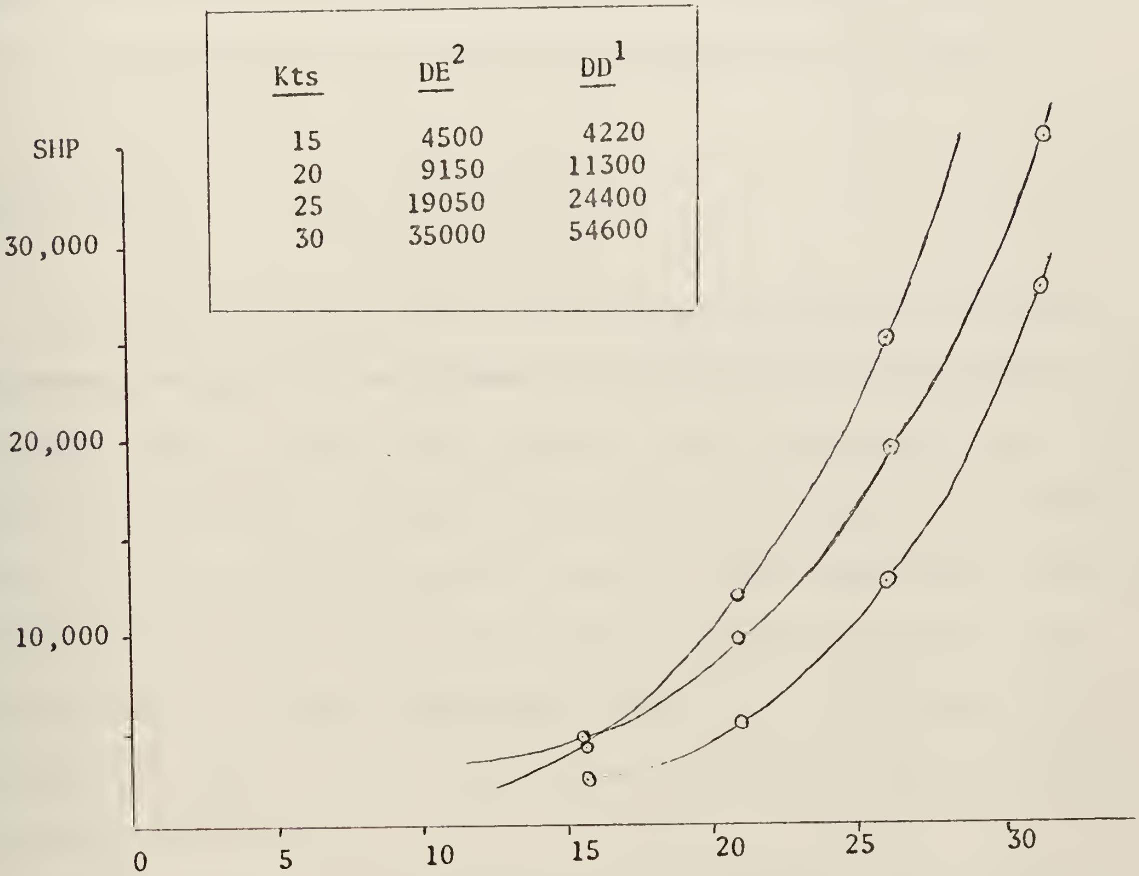



Figure 1. Speed Profile

## Power Requirements

Having an assumed speed profile, it is now necessary to find the power levels at these speeds to determine the design power levels of the propulsion cycle. Figure 2 was compiled using turbine and gear instruction books for applicable plants.




| Kts | DE[2] | DD[1] |
|---|---|---|
| 15 | 4500 | 4220 |
| 20 | 9150 | 11300 |
| 25 | 19050 | 24400 |
| 30 | 35000 | 54600 |

Figure 2. Power Verses Speed in Knots Curve

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[1] Turbine and Gears Instruction Book for the DD - 936, (NavShips 341-1301, GEI - 56786).

[2] Turbine and Gears Instruction Book for the DE/DEG, (NavShips 0941-011-8010, GEI - 60465).

These curves show that a power plant for a destroyer type ship should reach its peak efficiency at about 9000 SHP corresponding roughly to 20 knots in modern destroyer hulls. The peak power level should be about 35,000 SHP to be competitive with existing plants.

The electric power requirements were set at 1000KW for the normal underway load with an installed capability of 4000KW in at least two separate units for redundancy. This decision is discussed in the electric power section but is mentioned here for continuity of discussion.

## Cycle Choice

The propulsion cycles open to choice after satisfying the above requirements are very limited. There is the conventional steam plant currently used by the navy with a specific fuel rate of about 0.65 at this power level or the more advanced marine plants with specific fuel rates of about 0.44 if it has five feedwater heaters. A more complicated plant than the five heater plant is the reheat steam plant where the steam is extracted from the turbine, reheated and piped back to the turbine again. This plant is capable of fuel rates down to 0.41 but is too complicated to be feasible for navy use.

Another choice is the straight gas turbine engine. Until recently, this cycle has suffered from a poor fuel rate. Currently the most economical of these is the second generation LM-2500 aircraft engine manufactured by General Electric and slated for use in the DD 963 class of destroyers and the PF patrol boats. This engine has a very admirable fuel rate at full power of 0.39 (propulsive). At 9000 SHP this becomes 0.51 (propulsive);

and in order to compare it with the cycle for this thesis, it has to be corrected to an all purpose fuel rate. This figure then becomes about 0.60 when corrected for electric power and evaporator requirements. This figure is a little better than the current navy plants but worse than the merchant ship plants. Even more disquieting is what happens when two of these engines are run at the 20 knot power level which is what can be expected in the currently planned installations except when economy is the overriding consideration. Here the propulsive fuel rate becomes about 0.71 and the all purpose fuel rate about 0.91. A current navy plant runs about 0.82 at this power level and the merchant ship plant is considerably better.[3] These figures are summarized in graph form in appendix A.

Another alternative cycle is the combined diesel and gas turbine. This plant is hard to find fault with other than it requires some form of clutching with the associated problems of large clutches. Diesel engines of the size required are also large and have a problem with low frequency noise. This cycle is currently in use on the U. S. Coast Guard Hamilton class cutters.

Still another alternative cycle is the combined gas turbine and gas turbine cycle. One turbine would be small for cruise power and one large for full power. This system allows the gas turbine to operate at its best level most of the time eliminating the fuel rate problems. The clutches can also be eliminated by accepting the windmilling losses of the idle turbine. This cycle has also been used in some installations.

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[3] Comparative All Purpose Fuel Rates Chart, prepared by the ship propulsion section of General Electric Company, MST division, January 1971.

By far the most interesting and challenging cycle considered for this application is the combined steam and gas turbine cycle. This cycle has been manufactured for many years for land use and has been labeled a STAG cycle (Steam Turbine and Gas Turbine). In each installation to date this cycle has used the steam plant for base load with a large gas turbine for peaking power. This particular combination offers no advantage for a marine power plant as it would be easier to provide more or larger boilers to begin with if more power were desired.

A variation of the STAG cycle is a gas turbine with a heat recovery steam generator. This plant uses the heat which would be wasted in the gas turbine exhaust to generate steam which can be used for propulsion, hotel services or auxiliary loads. This plant, too, has been manufactured for many years both with fired and unfired boilers.

A marine application of this waste heat recovery cycle was the GTS JOHN SARGENT which was placed in service in 1956 and run for some 9700 hours. In this application the gas turbine provided all the propulsion power and the steam was used for powering auxiliaries and for providing hotel services. There was no supplemental firing.[4] Several other marine applications of this cycle are being designed but none are using supplemental firing.

The cycle analyzed in this thesis is one ideally suited to the destroyer application. There is a gas turbine for cruising supplemented

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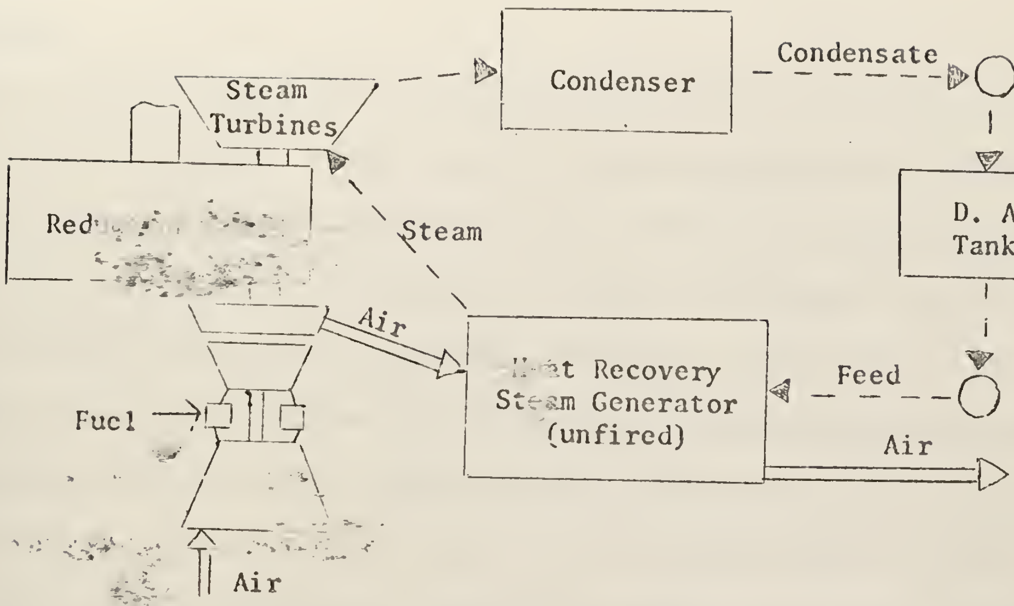
[4]Personal discussions with William H. VanCott, Marine Consultant under contract to General Electric Company (Lynn, Mass., June 1971 - August 1971). Mr. VanCott's prior experience includes almost 30 years at sea including duty as first engineer of the SS United States and chief engineer of the GTS John Sargent.

by a steam turbine using heat recovered from the gas turbine exhaust. In the cruise configuration there is no supplemental firing of the heat recovery steam generator. This is one design point for the boiler and where it should have its highest efficiency. As the power requirements increase, the boiler is fired as required. Ideally one would use a grid type burner and heat the exhaust gas from the gas turbine just before it passes through the boiler. The analysis in appendix F shows the power attainable under these conditions. The current state of the art of grid burners does not allow for this. They fire well only using natural gas and foul using heavy distillate or residual fuels. Therefore, the boiler in this study had to use a forced draft blower and conventional burners. This question is addressed again in the boiler section.

This cycle uses many simplification features and many efficiency improvers such as: variable steam pressure, controllable and reversible propeller, variable geometry gas turbine, constant inlet area steam turbine and an attached generator. It also has many reliability improvements, the greatest of which is the boiler. It is always hot as there is always hot gas passing through it. When it is fired, it is fired by a distillate fuel which virtually eliminates fireside maintenance. Figure 3 shows the cycle diagram at low and high power level in greatly simplified form.

Up to this point only the basic cycle has been outlined. Now the component selection must be made. Once again it must be stated that these components are not always the ideal ones but are those which come closest to ideal in keeping with the stated objective of using existing equipment or, at worst, readily redesigned and manufactured equipment. Further optimization is discussed in each section.

Cruise Power Level




Full Power Level

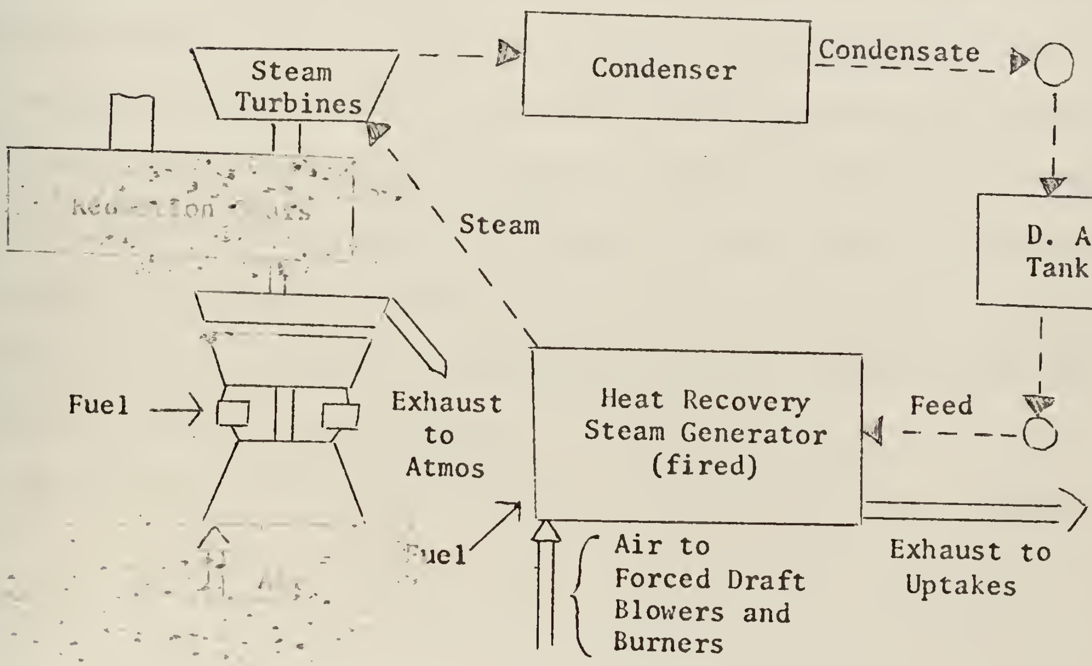



Figure 3. Cruise and Full Power Cycle Configurations

## Component Selection

### Gas Turbine

The single most complicated and difficult to design component in the cycle is the gas turbine. There are many turbines available off-the-shelf designed for many different applications with the majority being designed for aircraft use. All the aircraft gas turbines are designed with frontal area and weight minimization of paramount importance. Only the second generation LM-2500 has combined this weight and volume minimization with an outstandingly good fuel rate. This particular gas turbine has two disadvantages for use in this cycle. First, it has too much power to operate efficiently at the required power level, and secondly, it has the typical overhaul cycle discussed below.

All aircraft gas turbines have two common severe limitations when applied to marine plants. These are the short time between overhauls and the poor part load fuel rate. The overhaul cycle problems arise because all these turbines operate at the limits of current technology in temperature and pressure ratios in order to reduce size and weight. Consequently the overhaul interval is typically 10,000 hours of operation.[5] The part load efficiency problems can be partially overcome by using variable area geometry at the power turbine inlet.

By contrast, the overhaul interval of the industrial gas turbines is

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[5] J. V. Shannon et al., "DX Engineering Plant Life Cycle Cost Comparison Basic Steam Plant Vs. Basic Gas Turbine Plant Arrangement (GT - 1), Marine Turbine and Gear Dept., General Electric Company (Lynn, Mass., 16 October 1968).

typically 1000,000 hours of operation.[6] This is primarily due to a more conservative design approach using much less critical parameters and much less exotic metals.[7] One does pay a severe penalty in machinery size and weight because of this, but these industrial gas turbines still sell because the size and weight does not matter in a land plant installation.

In a marine plant one would ideally want a turbine designed for long hours of continuous use with an overhaul as long as or longer than that for the rest of the ship. This can be satisfied only by an industrial gas turbine. But, in a marine plant one also wants minimum space and quite often minimum weight as well. This dictates an aircraft gas turbine. The trade off on which way to go depends on the type of ship the installation is for. In a supertanker neither weight or space is very critical. In a container ship space is critical but weight is not. In a naval destroyer both are very critical which dictates using an aircraft gas turbine unless there is another way out.

An industrial gas turbine which is small enough to fit into the general size and weight limits imposed on a naval destroyer has been selected for this plant. This particular turbine also gives adequate power for cruise when waste heat recovery is employed. The turbine is the General Electric series 1000 which is currently designed for about 4900 SHP output. Some of the design parameters are listed in Table 1 which is

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[6] G. A. Ludwig, "Marinization of Industrial Heavy-Duty Gas Turbines," General Electric Gas Turbine State of the Art Engineering Seminar (SOA-17-71), June, 1971.

[7] A. O. White, "General Electric Heavy-Duty Gas Turbines," General Electric Gas Turbine State of the Art Engineering Seminar (SOA-16-71), June, 1971.

extracted from references 6 and 9, parts of which are included in appendix B.

General Electric Model 1000 Gas Turbine

| | | | |
|---|---|---|---|
| Number of Stages: | Compressor | | 15 |
| | Turbine | | 2 |
| Shaft Speed: | Gas Generator | | 10920 rpm |
| | Power Turbine | | 10920 rpm |
| Aux. Power Req. | A.C. Lube Oil Pump | | 15 HP |
| | D.C. Emerg. L. O. Pump | | 5 HP |
| | A.C. Ignition | | 500 watts |
| | D.C. Control | | 125 watts |
| | Starter, 450 VAC | | 100 HP |
| Rotation: | CW when facing load end | | |
| Performance on Distillate Fuel: | Air | 161,700 | Lb/Hr |
| | s.f.c. | 0.5668 | Lb/SHP-Hr |
| Weight on foundation with accessories: | | | 18600 Lbs |

Table 1.

In addition to this information, various conversations with Mr. A. O. White, Manager of Marine Products of the General Electric Gas Turbine Department, indicated the following possible improvements. This turbine could have its power output and air flow increased by 20% by a modification called "zero staging." This consists of adding one more stage of compressor blading ahead of the existing blading changing the output to that listed in Table 2. The fuel rate would remain about the same.

| | | |
|---|---|---|
| SHP | 6000 | |
| Air Flow | 194000 | Lb/Hr |

Table 2.

Mr. White also indicated that a substantial weight reduction would be possible for a naval design, but this reduction is not used at this time since it would require a substantial redesign and since firm estimates of the weight reduction were not possible.

This turbine is well suited to this application if "zero staged." Due to the subsequent requirement of using forced draft blowers at full power, one could use this turbine even without the modification because the extra performance and gas flow is not required. This would decrease the overall plant output by 20% which would increase the specific fuel rate about 30%. In addition to being well suited to use in this cycle, all the performance data is available at design and for off design and all the dimensions and weights are available. This eliminates a great deal of estimating and makes a determination of feasibility more believable.

## Steam Turbine

The steam turbine design for this cycle includes some features not usually included in a steam turbine because of the need to match it to the gas turbine. It must run at a constant speed over its entire power range. This is because the gas turbine is limited to a 2% overspeed to its alarm point and only 10% overspeed to its failure point.[8]

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[8] A telephone conversation with Mr. A. O. White, Manager, Marine Projects for the General Electric Company Gas Turbine Division, 9 July 1971.

The heat balance in appendix F shows that the power level at the cruise design point is about 3200 SHP. This is the point where the gas turbine is at full power and the boiler is unfired. From this point, the steam turbine must go to its maximum output of about 36,000 SHP.

Current technology limits on metal, pumps, etc. limit steam pressure to about 1450 psig. Steam plants of higher pressure are being considered but none have been installed in marine plants. The pressure at full power in this cycle was set at 1200 psig. This was mainly because that is the pressure currently used by the U. S. Navy. The temperature was similiarly limited to 950°F. Since this temperature cannot be reached in the unfired mode because of limitations on heat exchanger size, the steam temperature must vary. This raised the question of looking at variable steam pressure also.

Variable steam pressure is very well suited to the steam turbine design and allows a constant area inlet for most of the power range. The steam pressure was allowed to vary from 400 psig at the cruise mode to 1200 psig when at full power. These pressures were compatable with the boiler design and allow sufficient pressure at low powers to run all the required auxiliary equipment. The low limit on the pressure was not completely compatable with the steam turbine design and required the inlet area to be halved at the cruise mode. The basic idea is shown in Figure 4. This still eliminates the need for the complicated inlet nozzles present in all standard steam turbines. This modification is only a casing modification and can be easily accomplished. The power output of the turbine is controlled by the modulating valve until it is wide open at 400 psig. Thereafter the power is controlled by varying the steam pressure. If a

faster response is required, a full 1200 psig could be carried on the boiler and the power controlled by the modulating valve over the entire range, but this sacrifices efficiency.

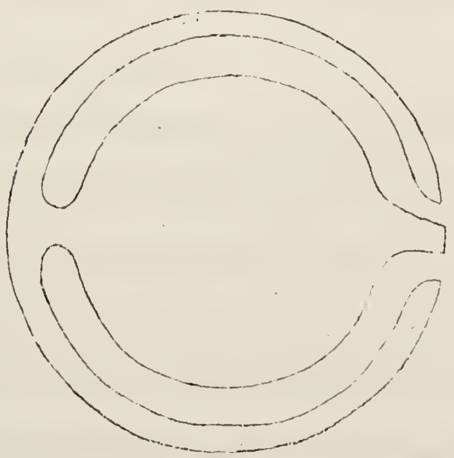


Figure 4. H. P. Steam Turbine Inlet Arrangement

The choice of operating temperature was based on the boiler design where a temperature of about 750°F at 400 psig and 950°F at 1200 psig was attainable from the same boiler. This is mentioned here because these temperatures are necessary to estimate the turbine efficiency.

The steam turbine efficiency was estimated using established procedures and then backing out the corrections for astern loss (there is no astern element), speed loss (constant speed) and part load loss (variable pressure).[9] The steam turbine has only exhaust loss when using constant inlet area, constant speed and variable pressure. This is a very strong

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[9] "Marine Steam Power Plant State of the Art Seminar, 1970," Babcock & Wilcox and General Electric Company (eds.).

argument for this cycle configuration. The estimate for steam turbine efficiency is shown in appendix C and is 0.790 at the cruise point and 0.824 at the full power point. It should be noted here that these figures are not as accurate as shown, but they are accurate to within about 2%. This is because this turbine has not been analyzed in sufficient detail to ensure more accuracy. However, this accuracy is more than adequate for this feasibility analysis.

The decision of whether to use a single cylinder or a multi cylinder turbine is primarily a function of the exhaust pressure and the steam rate combined with the room available for the turbines. In other words, it is a function of how much exhaust loss is acceptable. In order to keep the condenser size and weight down, a decision was made to accept 5 inches of mercury pressure at full power. This was justified by the fact that at cruise power this would be 1.1 inches and there would be little lost efficiency. A further justification was the little time actually spent at the higher power levels. This decision forced the use of a cross-compound turbine which had already been recommended because of the power level.[10]

## Heat Recovery Steam Generator

Preliminary calculations on the steam flow possible were made using procedures established by Sheldon and Todd and are shown in appendix F.[11] Extended talks with Mr. Sheldon in late July, 1971, showed that the cycle

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[10] Conversation with Mr. M. A. Prohl, Manager, Turbine Engineering, Marine Turbine and Gear Department, General Electric Company, August, 1971.

[11] R. C. Sheldon and D. M. Todd, "Optimization of the Gas Turbine Exhaust Heat Recovery System," ASME Paper 71-GT-79.

was possible although the feasibility of a boiler for it was questionable. Further research in this area proved that a convective heat transfer boiler could not satisfy both the cruise and full power requirements. Therefore, the boiler has to use radiant heat transfer at full power.

It should be pointed out here that boiler design is not a science but more of an art. No one to date has been able to explain exactly what happens inside a boiler, but they can come close in boilers of a conventional design. When the design is like the one for this cycle where there is no radiant heat transfer in one mode and there is in the other mode, the calculations must be made by hand. These calculations require the emperical formulas and charts developed over many years of boiler design. Even then the calculations are an iterative process.

The boiler design calculations are included in appendix D. These calculations were made by Mr. Carl Horlitz of Combustion Engineering, Inc. and by the author of this thesis. They are accurate to within 10% which is sufficient for the purposes of this feasibility study. This, in fact, is about the degree of accuracy attainable in a boiler design of this type. A more accurate study can only be made on a boiler of conventional design where the parameters are fairly well known to each design.

The boiler selected for this cycle is the Combustion Engineering, Inc. boiler designed for use in the AOE class ships for the U. S. Navy. The reasons for this choice were many. First, it is all ready designed to navy standards and this cycle is applicable to navy ships. Second, it has the required full power flow rate. It is easily upgraded to the 1200 psig required for full power. It was then necessary to calculate whether or not this boiler would fulfill the cruise requirements.

As can be seen in appendix D, the boiler does fit the cruise requirements, but would require a slight modification to the economizer. It also required a modification to the air flow path. The air would enter through the floor of the boiler rather than the front in the cruise mode. This modification was made to reduce the pressure drop through the boiler and retain the maximum gas turbine performance possible.

Another change had to be made from the originally conceived design in that the gas turbine exhaust has to bypass the boiler at full power. This is due to the high pressure losses in the conventional burners and the fact that there simply is not sufficient oxygen available in the exhaust gas to fire the boiler to the required full power level. If grid burners (burners consisting of a patchwork of pipes with holes on the downstream side which provide a large but low density flame area) could be made to burn heavy distillate fuel well, this type of burner could be placed in the gas path at the bottom of the boiler. This would allow a steam rate which would be considerably below the 230,000 Lb/Hr desired. But this type of firing would be sufficient for almost all the underway speed requirements of a destroyer type ship as shown in the assumed speed profile. If grid burners were satisfactory, they are what should be installed. One could then use a supplemental forced draft blower and burner for the final boost.

This requirement to bypass the gas turbine exhaust in the fired mode adds considerable complexity to the system in the form of additional ducting, a bypass valve, and much larger forced draft blowers than originally conceived. This is an unfortunate necessity forced upon the cycle because of the state of the art of grid burners.

The results of the boiler calculations are summarized in Table 3. The full power steam temperature is just a little too high and should be

lowered to 950°F. This adjustment was made for the heat balance calculations. This is a minor adjustment and could be made by removing some of the superheater surface area or using an attemperator which is the current practice in merchant plants where a constant superheater outlet temperature is desired. The heat balance also showed that some desuperheated steam was required in the cruise mode. This adjustment was also made during heat balance calculations.

Summary of Boiler Calculations

| | | Cruise Power | | Full Power | |
|---|---|---|---|---|---|
| | | Blr Calc | Heat Bal | Blr Calc | Heat Bal |
| Air: | Firebox | 950°F | - - | 2752°F | - - |
| | After Superheater | 816°F | - - | 1757°F | - - |
| | After Main Bank | 492°F | - - | 761°F | - - |
| | Stack | 364°F | - - | 355°F | - - |
| Steam and Water: | | | | | |
| | To Economizer | 280°F | 280°F | 280°F | 280°F |
| | | 415 psig | 415 psig | 1275 psig | 1275 psig |
| | To Main Bank | 448°F | - - | 407°F | - - |
| | | 415 psig | - - | 1275 psig | - - |
| | In Steam Drum | 448°F | - - | 578°F | - - |
| | | 415 psig | - - | 1275 psig | - - |
| | Superheater Outlet | 758°F | 758°F | 997°F | 955°F |
| | | 400 psig | 400 psig | 1200 psig | 1200 psig |
| | Desuperheater Outlet | - - | 456°F | 655°F | 655°F |
| | | - - | 395 psig | 1160 psig | 1160 psig |
| Flows: | Air (Lb/Hr) | 194000 | 194000 | 293238 | 293238 |
| | Superheated Stm (Lb/Hr) | 25000 | 22250 | 230000 | 210000 |
| | Desuperheated Stm (Lb/Hr) | 0 | 4370 | 30000 | 31600 |
| Draft Losses (In. Water) | | 4.36 | - - | 48 | - - |

Table 3.

The extremely low draft loss at the cruise power is noteworthy. This loss allows the gas turbine to operate at very near its design point in this mode. At full power the draft loss is considerably higher and reflects the burner losses rather than any great increase in the boiler loss. The draft loss through the boiler if the burner loss is neglected is about 15 inches of water.

Figure 5 is a diagram of the air flow arrangement of the boiler. The gas bypass and the reasoning for the floor located air inlet has been explained. The ducting immediately below the boiler must be brick lined as shown because of the intense heat in the firebox when the boiler is being fired.

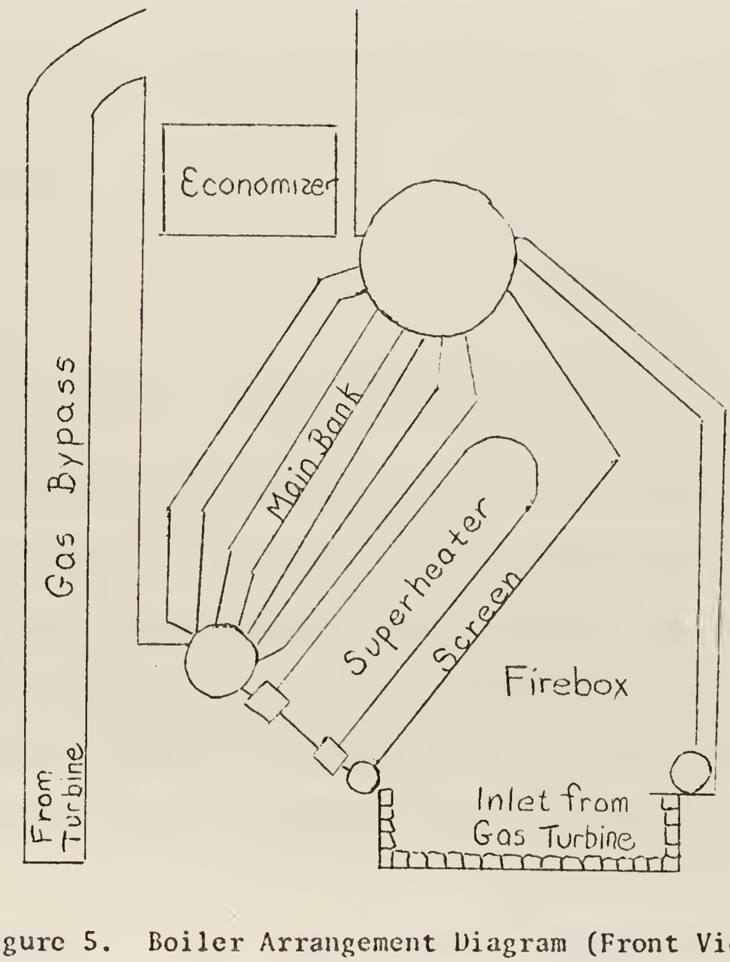


Figure 5. Boiler Arrangement Diagram (Front View)

Figure 6 shows a schematic of the ducting for the exhaust gas routing system. This damper is arranged to minimize possible damage to the gas turbine due to inadvertant shutting or flareback. The two dampers would be connected mechanically to ensure that one or the other is always open. They are hinged at the upstream end so that the gas flow would keep them open should the mechanism fail.

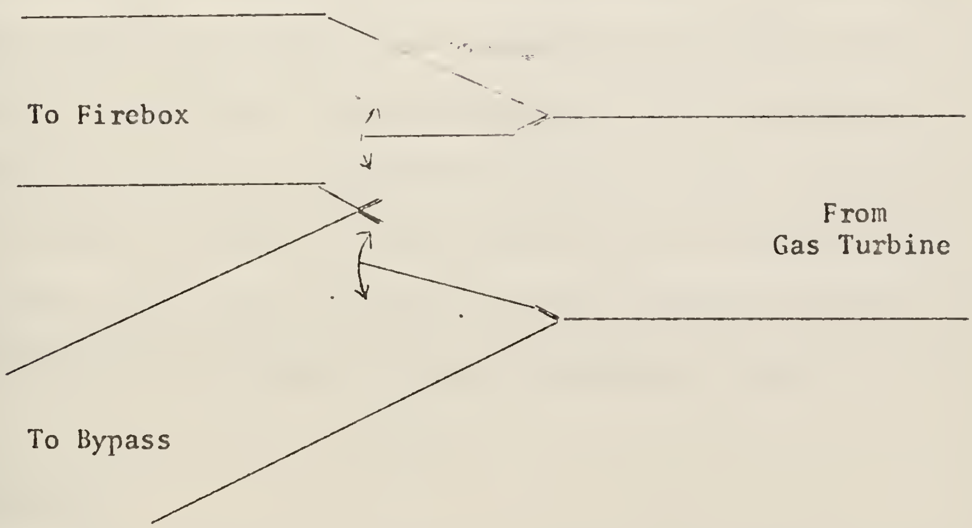


Figure 6. Air Flow Ducting Schematic

The boiler bypass could be routed to either the inlet or the outlet side of the economizer. Because the full power mode where there would be a full bypass of gas turbine exhaust gases would be used so seldom, the total effect of routing these gases to either location would be minimal. Therefore, they were routed to the outlet side of the economizer to keep exhaust losses down on the gas turbine and to keep the total height of the boiler down.

Using the configuration of this proposal, the reaction time to raise

the boiler from the cruise condition to the full power condition would be on the order of 7 to 8 minutes. If a faster response time were needed, the controls could be arranged to keep the boiler at 1200 psig at all times. This would reduce the reaction time to about 2 to 3 minutes. But it would also reduce the cruising efficiency of the plant by requiring a less efficient steam turbine. It would also reduce the plant maintainability by requiring much more stringent feedwater requirements. It would also reduce the boiler reliability somewhat due to continued operation at the higher temperatures and pressures. The reaction time of 7 to 8 minutes does not seem unduely long when one considers the number of times that maximum acceleration would be desired. There could even be an emergency mode on the control system which would allow this constant pressure operation when desired since the steam turbine does have a modulating valve.

## Reduction Gears

The reduction gears chosen for the cycle are a straight forward, naval designed, double reduction, locked train type much like those installed at present. These gears allow the spread required to fit in the cross-compound steam turbine and the gas turbine. This required width ruled out planetary gears. Figure 7 shows a typical set of reduction gears for a destroyer type ship. The set required for this cycle would add an input on the after end for the high pressure steam turbine and a take-off for the attached generator (discussed later).

Gugliuzza and Hargett give sample drawings of several gear arrangements one of which is for a COGAS cycle designed to fit into a DD-931 class destroyer. This particular cycle used five input shafts with an articulated

gear train for all inputs instead of the conventional locked train type. The reason for this departure from normal was due to the number of gears they had to cluster around the bull gear. The cycle proposed here needs only two outputs or inputs on each end and, therefore, should easily fit a conventional design. Further discussion by Smith on the state of the art of gear design leaves little doubt that this is possible.[12]

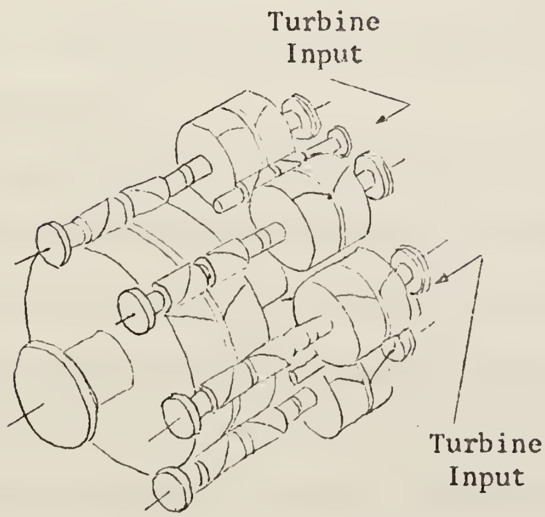


Figure 7. Typical Destroyer Gear Set[13]

Propeller

The choice of the proper propeller for this cycle is restricted to controllable and reversible pitch propellers because of the constant speed requirement of the gas turbine over the entire upper range of power. At

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[12] "Marine Steam Power Plant State of the Art Seminar, 1970," Babcock and Wilcox and General Electric Company (eds.).

[13] T. A. Gugliuzza and W. H. Hargett, "Gear Design and Laboratory Test Experience--Marine Turbine Propulsion," ASME Paper 69 - GT3.

the lower power levels of the gas turbine, such as for maneuvering, more power is available faster using the constant speed control, so the C & RP propeller is still better.

The controllable and reversible propeller offers a flexibility unmatched by any fixed pitch propeller and can be programmed to give whatever pitch one desires at any power level and rpm. This leaves the final choice of control sequence open to any variations required by the particular application of the cycle. The curves included in appendix E show just how versitile these propellers are.

The decision on controlling the propeller for this cycle was made after a discussion with the Bird-Johnson Company of Walpole, Massachusetts. At the power levels of this cycle there could be a strong cavitation signature if the choice is a constant speed shaft, but this type of control is entirely possible. In this same discussion it was indicated that this signature problem should be overcome in the not-too-distant future. Since constant speed offers several efficiency advantages, the decision was made to use it. The two major improvements of constant speed on the shaft are: a greatly improved steam turbine efficiency at part load and the ability to use an attached generator for normal cruising electric power. The turbine efficiency improvement factor can be seen in appendix C and is about 0.85 at half speed. The improvement by using an attached generator is discussed in the following section.

Appendix E has a line drawing of a BLH controllable pitch propeller designed for use in naval destroyers of about this horsepower. It shows the hub and control unit mechanics. Also included in this appendix are typical control sequences for this propeller.

There is much controversy over whether one loses or gains efficiency with controllable pitch propellers. The further one pursues this question, the more it becomes obvious that the answer depends on who is giving the opinion. One school of thought insists that the larger hub and general purpose design of the individual blades cause a loss of propulsive efficiency. The other school of thought is that, while there is some loss of efficiency at the design point, the C & RP propeller is more efficient at off design and reverse points. Therefore, for a naval destroyer, which operates over a wide range of speeds continuously, the C & RP propeller appears to have a significant advantage. In addition, it allows for much faster response in a maneuvering situation.

## Electric Power Generation

The last major decision to be made prior to conducting a heat balance is that of electric power generation. The choices here consist of steam, gas turbine, or diesel powered or attached generators. In recent studies, the diesel generator has been ruled out because of its low frequency noise and maintenance problems.

By far the most efficient method of generating electric power is by means of an attached generator. For this method the main shaft must run at a fixed speed. The generator would be run off the reduction gear and, therefore, could not be very large because of space limitations in this cycle.

The next most efficient method is to use a steam turbine as the power source since there is already steam present in large enough quantities. The turbine could exhaust into the main condenser thereby using the greater

vacuum attainable there over an auxiliary condenser.

The third choice would be for a gas turbine power source. Here again all the pros and cons associated with industrial verses aircraft turbines arise. Since they were discussed previously, they will not be discussed here.

Because efficiency is one of the main goals of this cycle, the decision was made to go with an attached generator large enough to carry the normal underway electric load. This will make it small enough to fit aft of the reduction gear beside the main shaft. The size of this generator was set at 500 KW on each shaft.

Naval destroyers often require much greater powers when running all the combat equipment. But these requirements are only for short durations under normal conditions although they could be for extended time in combat. Therefore, the decision was to use an industrial gas turbine power source. These generators would be instantly available for peaking loads and could also be run for extended times when required due to the industrial design.

The particular choice for this cycle was the AVCO TF25A Industrial design gas turbine which will run on navy distillate fuel. This engine develops about 1500 KW continuously and 1650 KW maximum at a specific fuel rate of about 0.693 Lb/SHP-Hr or 0.924 Lb/KW-Hr. These are the numbers used in the heat balance at full power where additional power is required to run the electric pumps in the propulsion plant.

The 1500 KW power level was chosen to place the total installed electric power capability at 4000 KW. In a two shaft ship using two of these generators plus two attached generators this number is achieved.

There could be a problem with synchronization of the gas turbine and

attached generators if the ship were maneuvering violently. Therefore, in the maneuvering situation one would probably use the gas turbine generators.

## Auxiliaries

The various auxiliaries for this cycle were sized using the references listed in each case and integrating them with established guidelines.[14] All the specific pressures and steam flows are in the heat balance.

Main Condenser: This was designed by standard methods of reference 4. This item was sized in order to attain 5 inches of mercury absolute pressure at full power and then the vacuum at cruise power was calculated.

Forced Draft Blowers: Reference 7 was used to find the required steam flow at full power.

Feed Pumps: Reference 10 was used to establish the power requirements of the pumps. The decision was made to use a multiple speed electric motor for the power because of the variable pressure of the output and the greater efficiency of the electric motor over a steam pump.

Circulating Water Pump: This pump is an electric driven pump which is installed but not used underway above approximately 5 knots. It is, therefore, not included in the heat balance calculations.

## Heat Balance

The heat balance results are in appendix F. These results show that

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[14] S. T. Holm, "Recommended Practices for Preparing Marine Steam Power Plant Heat Balances," (revised in 1970 by C. W. Stott, Jr), SNAME Technical and Research Bulletin, No. 3-11a.

the cycle does in fact fit together. The fuel rate is not as good as originally expected or attainable with modifications. But the componants for the cycle were chosen with a naval application in mind and, therefore, some efficiency was sacrificed. Examples of these losses are an unusually large evaporator load, a greater than normal electric load, a lower than normal feedwater temperature, a cycle with only two feedwater heaters, and relatively inefficient forced draft fans.

| | Sheldon & Todd | Heat Balance |
|---|---|---|
| Steam Turbine Horsepower | 3100 | 3180 |
| Gas Turbine Horsepower | 5760 | 5760 |
| Total Horsepower | 8860 SHP | 8940 SHP |

Table 4.

## Weight Analysis

A summary of weights for this plant is shown in Table 5. In each case the reference or references are listed. Some degree of comparison can be made with the General Electric proposal for the DX project which had an installed equipment weight of 686,000 pounds. Standard guidelines for steam plant installations allow for much more than that.

When a propulsion plant weight comparison is being made, the total weight to be compared should include everything necessary to accomplish the same task. The electric loads should be comparable in that the power available to the ship after the propulsion plant power is subtracted should be

Summary of Propulsion Plant Installed

Equipment Weights

| | WT. (Lbs) | References |
|---|---|---|
| Main Propulsion Turbines and Gears | 148000 | 15 |
| Propulsion Gas Turbine with Accessories | 16000 | 9 |
| Lube Oil System (Cooler, Pumps, Strainers, Purifier, Heater) | 12000 | 15 |
| Feed Water System (Cruise Pump, Main Pump, Booster Pumps, DFT) | 42000 | 15 |
| Vacuum and Condensate System (Condenser, Vacuum Pump, Circulating Water Pump, Condensate Pumps, Gland Leakoff Cond., Gland Exhaust Fan, Drain Tank and Pump) | 88000 | 15 |
| Steam Generation System (Boiler, Controls, Fans) | 230000 | 15, 2 |
| Electric Power Generation (Attached Gen., Gas Turbine Generator) | 28000 | 1 |
| Miscellaneous | 14000 | 15 |
| TOTAL | 578000 Lbs | |

Table 5.

the same for each case. The evaporator load should be adjusted so that each case considered has the same number of gallons per man available after the propulsion plant requirements are met. A weight comparison should also include the weight of the fuel required to meet endurance specifications. The figures used to make the fuel calculation should be the all purpose fuel rate in each case. Table 6 shows a sample comparison of the fuel load to meet endurance. These figures are for a 7755 SHP per shaft power level

using attached generators at the 500 KW load. The electric load of the steam plant is met by steam powered generators at the 500 KW load.

If a lower power level were compared, the gas turbine plant would increase its fuel comsumption dramatically and the steam plant would increase to a lesser degree. The cycle analyzed in this thesis would increase only slightly.

Comparison of Cruise Fuel Load for Three Comparable Propulsion Plants

| | | |
|---|---|---|
| This Cycle (One shaft) | 0.438 | 333 tons |
| 1200 psi DD Plant (One shaft) [Current U.S.Navy practice] | 0.64 | 497 tons |
| LM-2500 (One operating, evaps incl.) | 0.57 | 437 tons |

Table 6.

## Volume Analysis

It is clear that the gas turbine plant has a considerable advantage over other propulsion plants as far as propulsion equipment is concerned. This advantage is reduced somewhat when the total plant volume is considered. However, the cycle proposed in this thesis does have an advantage over a conventional steam plant.

There is no second boiler. Even though the one boiler is large when

compared to typical destroyer boilers, it is small when compared to the two required in the steam plant installations. The total fireroom length would be reduced by about 8 feet in the DE-1052 Class. The engineroom, however, would have to be about the same length.

The auxiliary equipment space would be about the same volume as that in a conventional steam plant but would be reduced somewhat by the absence of auxiliary condensers and their associated equipment. The gas passage ducting could raise this again to have the volume about the same as that of a steam plant. This ducting would be routed down from the gas turbine and along the hull to the bottom of the boiler. In this manner the volume would be used where is normally wasted space in a conventional steam plant.

## Reliability

The reliability study was made using Prantis's publication as a guide.[15] The items not included in this reference were estimated using information made available by the manufacturer in some cases and simply guessed at in some other cases. These values are shown in Table 7. The mission time was taken at 225 hours or about the total endurance time of this plant. The full power reliability is: R = 0.983. The cruise power reliability is: R = 0.984. The "take home" power reliability is R = 0.999.

A part of reliability is redundancy. This is one area where this cycle is very good. The ship would be able to run on either the gas turbine or on

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[15] Edwin R. Prantis, "Tanker Steam Plant Reliability," Paper presented to the New England Section of SNAME, 15 January 1971.

the boiler alone. In a casualty situation they could be uncoupled and the remaining one used.

Reliability Data

| | $\lambda$ X $10^6$ (Failures per Million Hrs X $10^6$) |
|---|---|
| Boiler | 11.0 |
| Forced Draft Blower | 6.0 |
| Steam Turbine | 5.3 |
| Main Condenser | 3.0 |
| Main Condensate Pump | 6.0 |
| Main Vacuum Pump | 6.3 |
| Heat Exchanger and Accessories | 4.0 |
| Main Lube Oil Pump | 4.5 |
| Main Lube Oil Cooler | 2.0 |
| Fuel Oil Service Pump | 7.0 |
| Deaerating Feed Water Heater | 1.0 |
| Main Feed Booster Pump | 7.0 |
| Main Feed Pump | 8.0 |
| Gas Turbine | 14.0 |
| Adjustable Damper | 1.0 |
| Ship's Service Electrical System | 17.2 |
| Main Reduction Gear | 0.7 |
| Shaft, Bearings & C&RP Propeller | 2.6 |

Table 7.

## Discussion of Results

This thesis describes a feasibility study of a particular ship propulsion cycle. Each of the components was chosen using existing or easily modified and manufactured equipment. There are several areas where the efficiency and desirability of this cycle could be improved if the state of the art allowed. This is particularly true in the case of the grid burner and in the case of the auxiliary equipment as discussed in the heat balance section. In the case of the auxiliary equipment, further efficiency improvements could be made by accepting merchant practices rather than naval practices.

The weight and volume analysis is very dependent on the specific application of the cycle and the specific choices of various components. Therefore, no attempt was made to be any more accurate than necessary to show that these parameters are competitive with existing cycles.

One of the great advantages of this plant not discussed in the text is the ease of maintenance of the plant as a whole. The gas turbine should require very little maintenance other than the normal overhaul. The boiler spends almost all of its life cycle at pressures from 400 psi to 600 psi and is designed for 1200 psi operation. This should make boiler waterside maintenance minimal. The fireside maintenance should be similiarly reduced because the boiler is seldom fired; and when it is, it is fired with distillate fuel which virtually eliminates fireside deposits. The feedwater quality tolerances should be set for 1200 psi operation and, therefore, should further reduce waterside problems.

## Conclusions

This thesis has shown that this cycle is feasible which was the primary goal. Further, it is feasible to assemble the cycle using existing or readily available equipment.

The desirability of the cycle is much less obvious and requires some thought on the part of the reader after the first introduction to it. The desirability of the extremely good fuel rate is obvious. But the question of plant complexity appears to overpower the good fuel rate until some thought is given to it. The most complex question in the cycle is that of control. This study has assumed a constant speed propeller for reasons given in the text. There are many components in the cycle which have to be controlled separately and integrated to give smooth transition from cruise to full power.

The control of the cycle would be much easier if a grid burner were included as there would be very little mechanical manipulation to get from cruise power to about 70% power. Above this power level, there would have to be a boiler bypass in order to get the firebox hot enough. But up to that point the only control necessary would be to change propeller pitch and increase the fuel to the boiler. Both of these could be related directly to steam drum pressure.

The cycle appears to be quite complex at first glance. But, if one looks long enough at it, he can see that the complexity of the steam portion has been reduced greatly. The gas turbine is not a very complex machine, but is almost completely self contained and can be programmed to be controlled in a great variety of ways. In the case of this cycle, it would

remain at full speed as the power is reduced.

When considered as a whole cycle and all the advantages and disadvantages are considered, it appears that this cycle is worth considering for a prototype installation and for further study.

## Recommendations

This cycle should be applied to a very specific use and a complete study accomplished including the control question. Then a prototype plant should be assembled and run.

Should this cycle ever be considered for an actual shipboard installation, all the [illegible] should be designed and sized to that particular installation and every step should be taken to optimize the cruise fuel rate at the power level required for cruise. This study has taken the maximum cruise rate possible for its optimization.

## References


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10. Holm, S. T., "Recommended Practices for Preparing Marine Steam Power Plant Heat Balances," revised in 1970 by C. W. Stott, Jr., SNAME Technical and Research Bulletin No. 3-11a.
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15. "A Proposed Steam Turbine Propulsion Plant for the DX Project," General Electric Company, Steam Propulsion Group, Lynn, Mass.

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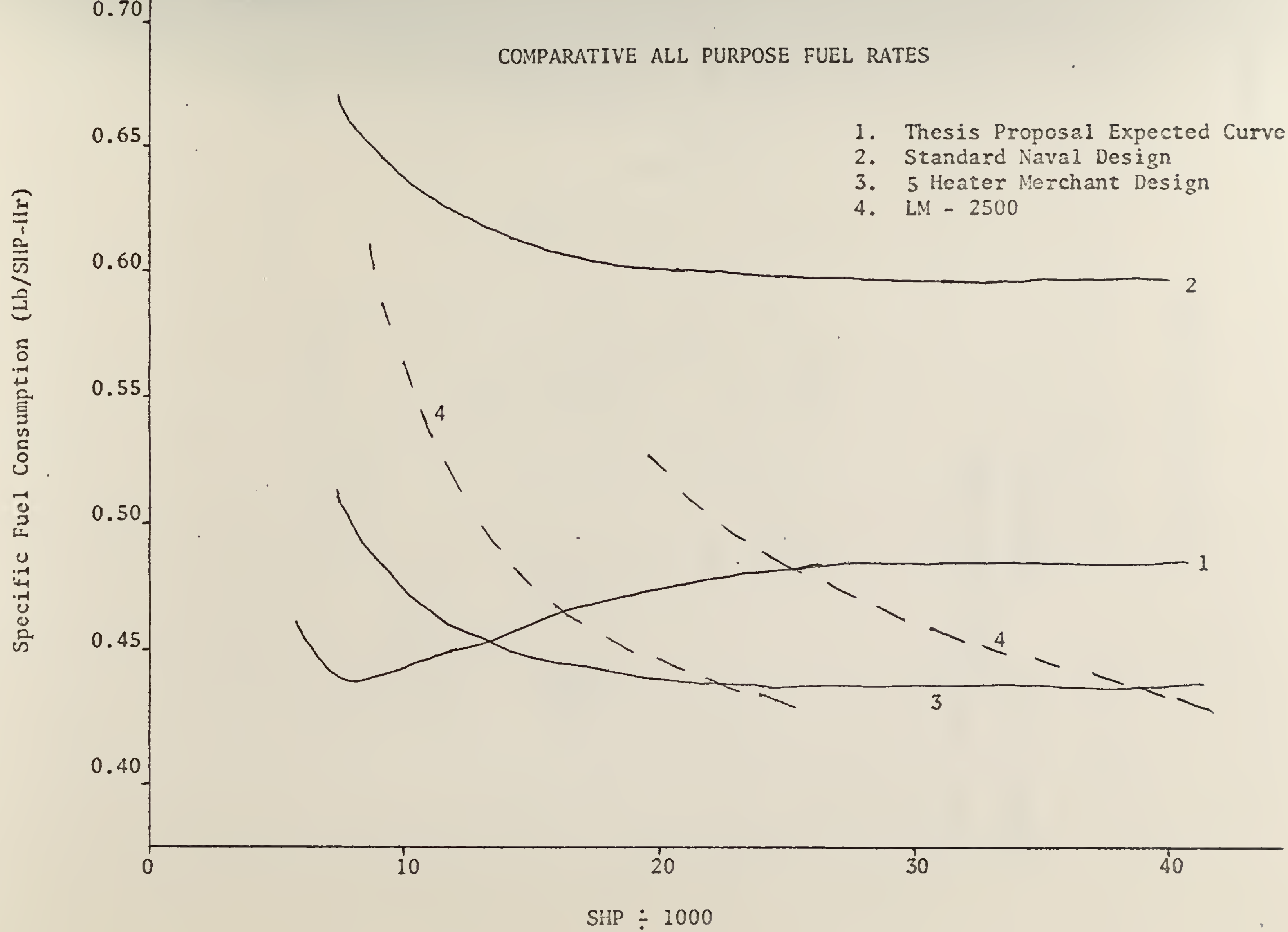
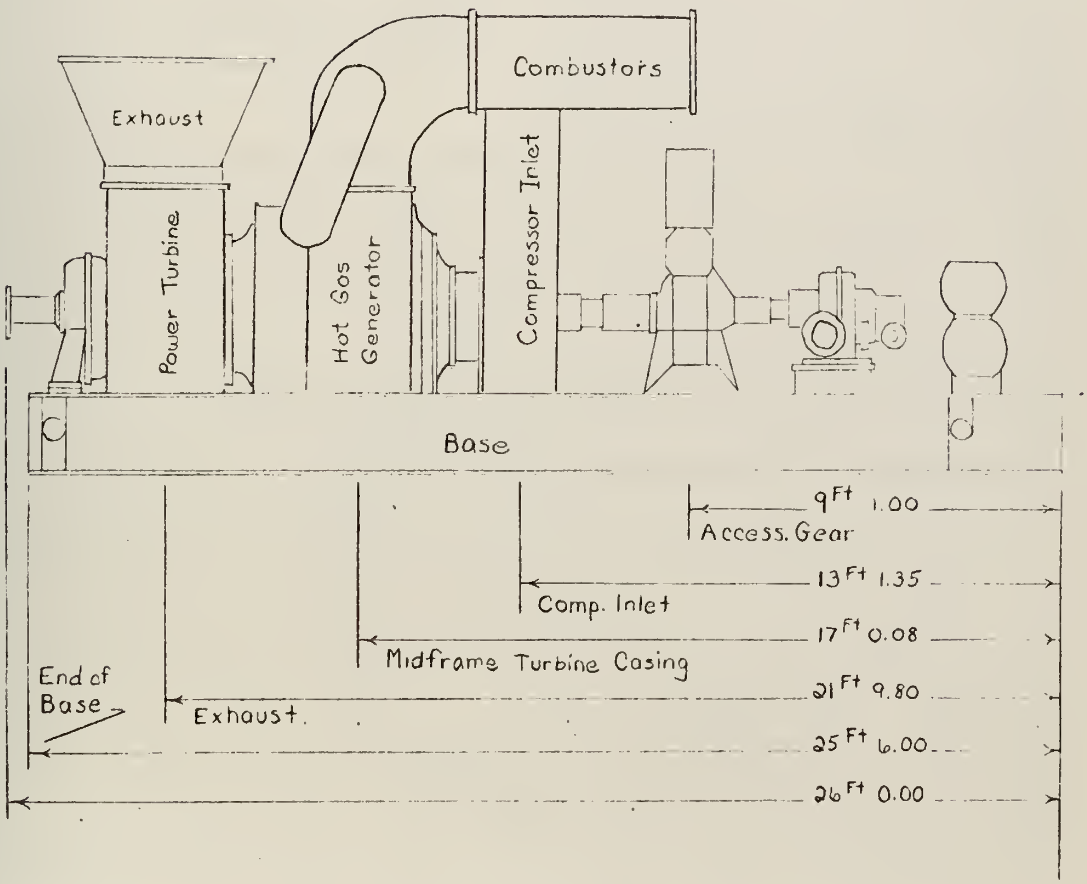
## Appendix A


COMPARATIVE ALL PURPOSE FUEL RATES
1. Thesis Proposal Expected Curve
2. Standard Naval Design
3. 5 Heater Merchant Design
4. LM - 2500
Specific Fuel Consumption (Lb/SHP-Hr)
0.70
0.65
0.60
0.55
0.50
0.45
0.40
0
10
20
30
40
SHP ÷ 1000
1
2
3
4
4

Appendix B

MODEL 1000 ELEVATION VIEW

| | |
|---|---|
| Max. Width | 10 FT 8 IN |
| Height to Top of Combustor | 11 FT 6 IN (approx) |
| Weights: Gas Turbine with Base and Accessories | 18,600 Lbs |

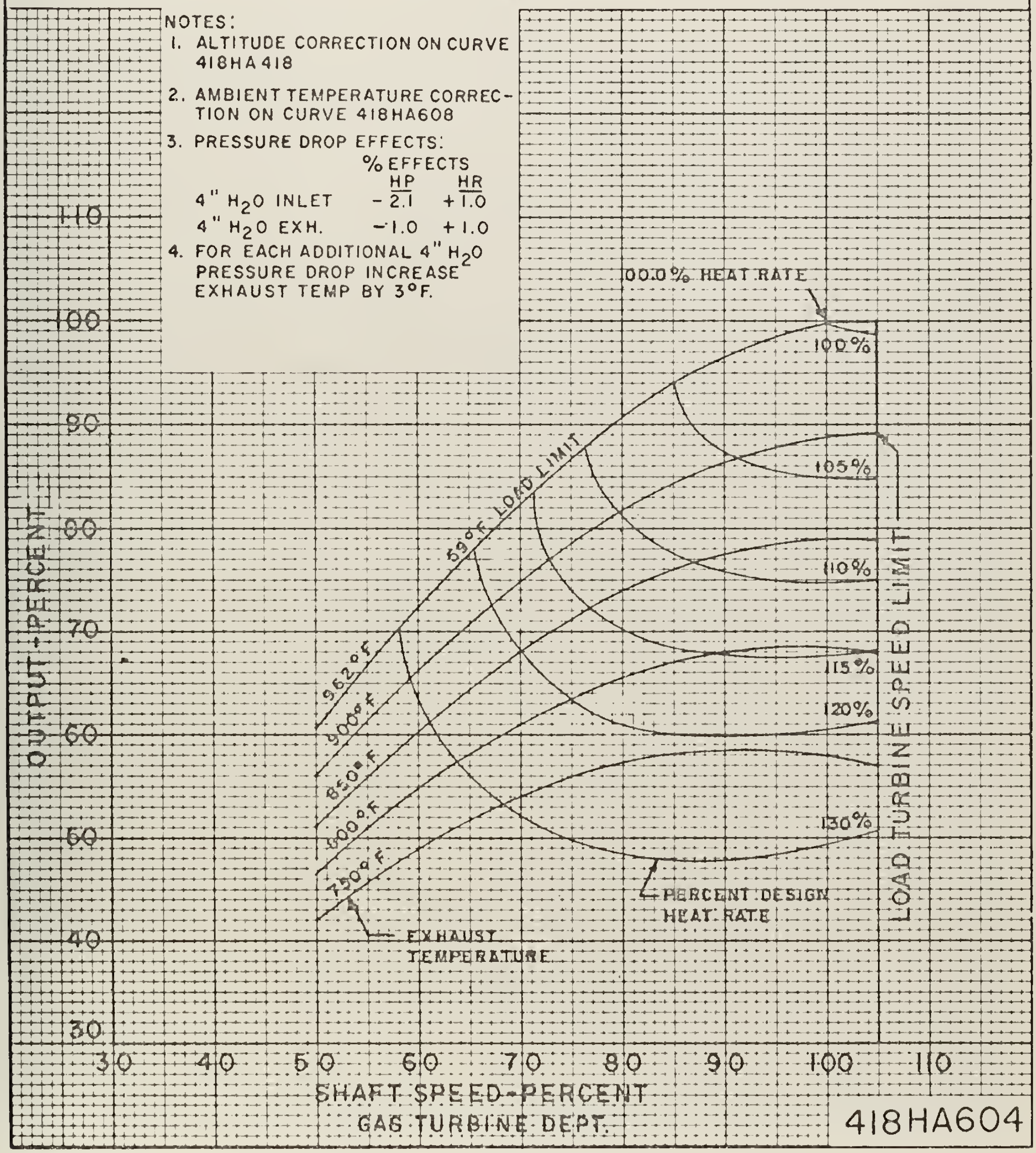


# GENERAL ELECTRIC MODEL M1502 *5050HP GAS TURBINE

ESTIMATED PERFORMANCE
COMPRESSOR INLET TEMPERATURE 59 F (15 C)
COMPRESSOR INLET PRESSURE 14.7 PSIA (760 mm of Hg)

| FUEL | | * NATURAL GAS | DISTILLATE OIL |
|---|---|---|---|
| DESIGN OUTPUT | HP | 5050 | 4900 |
| DESIGN HEAT RATE (LHV) | BTU/HP-HR | 10200 | 10430 |
| DESIGN FUEL CONSUMPTION (LHV) | BTU/HR | $51.5 \times 10^6$ | $51.1 \times 10^6$ |
| RATIO HHV/LHV | | 1.11 | 1.06 |

DESIGN AIR FLOW 161,700 LBS/HR
DESIGN SHAFT SPEED 10290 RPM




J. M. SILER DEC. 4, 1970

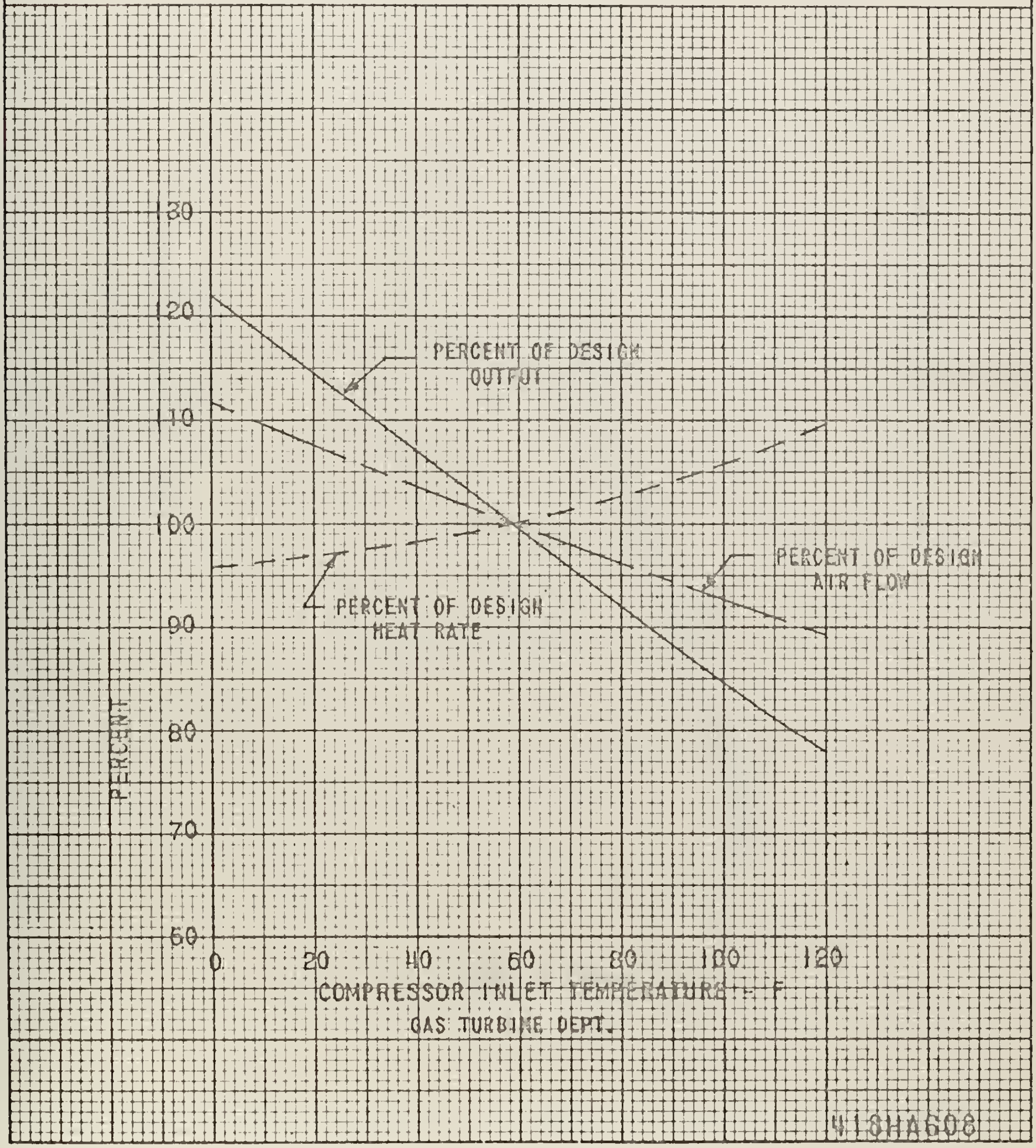
Appendix B

# GENERAL ELECTRIC MODEL SERIES 1502 GAS TURBINE

EFFECT OF COMPRESSOR INLET TEMPERATURE ON
MAXIMUM OUTPUT, HEAT RATE, AND AIR FLOW

100% SPEED

Curves for: 100% Compressor Speed
1675°F Firing Temperature

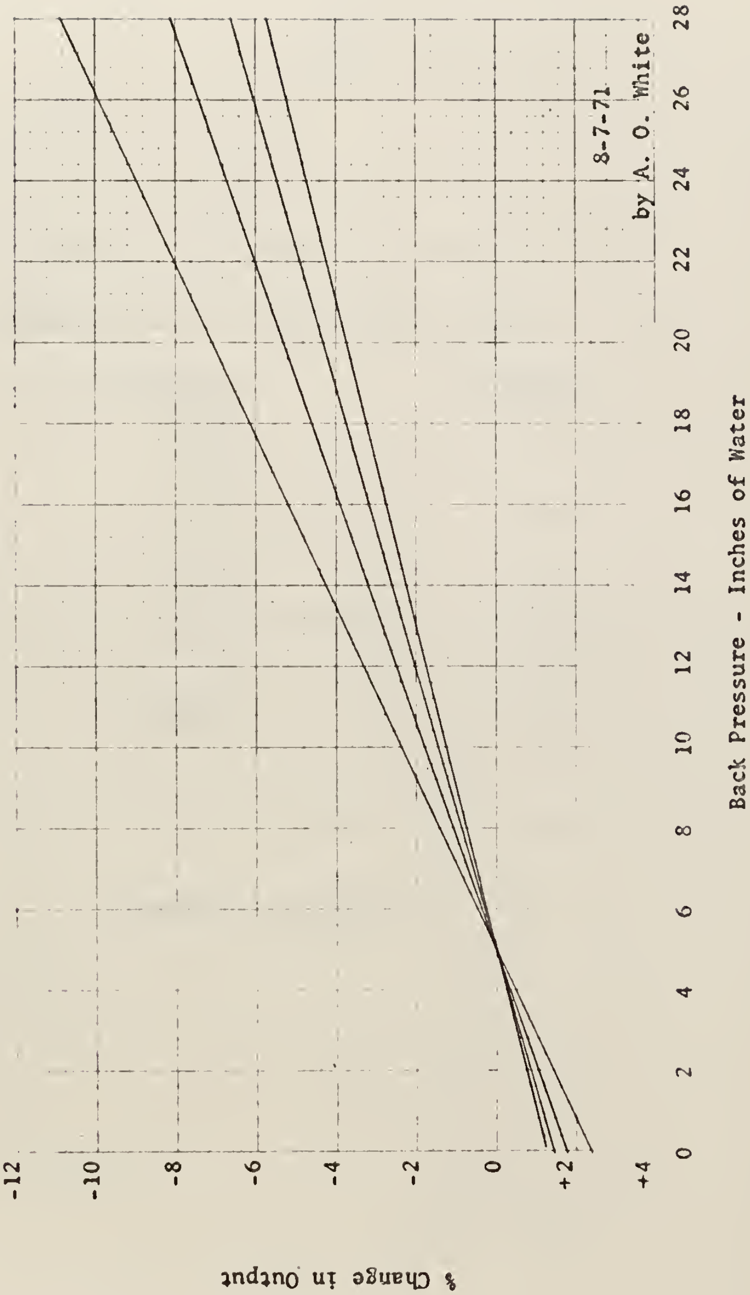


Appendix B

GAS TURBINE (Two-Shaft Cycle)

BACK PRESSURE CORRECTION

AT 70°F AMBIENT



Appendix B

## ANALYSIS OF INLET AIR

| | By Wt | |
|---|---|---|
| $O_2$ | 23% | $C_p = 0.24$ Btu/Lb |
| $N_2$ | 77% | 60% Relative Humidity<br>80°F |

## ANALYSIS OF EXHAUST AIR USING

## DISTILLATE OIL AT F/A RATIO 0.01308

| | By Volume | By Weight |
|---|---|---|
| $CO_2$ | 2.6% | 4.1% |
| $H_2O$ | 4.7% | 3.0% |
| $O_2$ | 16.2% | 18.1% |
| $N_2$ | 76.5% | 74.9% |
| $SO_2$ | 0.0034% | 0.0076% |

## ANALYSIS OF FUEL OIL

| | |
|---|---|
| API Gravity | 33° |
| Diesel Index | 45.5 |
| C/H Ratio | 6.7 |
| Sulphur Content | 0.3% |
| LHV | 18,350 Btu/Lb |

Extracted from: General Electric Gas Turbine Department PEI 4.1.1
12-10-62

Appendix C

## ESTIMATION OF STEAM TURBINE PERFORMANCE

(based on references 10 and 13)

| | Full Power | Cruise |
|---|---|---|
| Basic Turbine Efficiency (ref. 10) | 0.84 | 0.84* |
| Temperature Correction (ref. 10) | 1.02 | 0.99 |
| Exhaust Loss Factor (ref. 13)** | 0.95 | 0.96 |
| Estimated Turbine Efficiency | 0.824 | 0.790 |

*Reference 10 gives excellent "good practice" rules for calculating the efficiency but must be combined with what the author was told by Mr. Prohl.

**The calculation procedure in reference 13 is quite involved and complicated and must be corrected for various changes one would make for this cycle.

## SCOPE OF PERFORMANCE DATA

The performance tables in this section include combinations which could result in impractical designs. However, in order to include all of the combinations which it might be desirable to study, the scope selected is necessary.

## PROCEDURE

Steam rates at rated steam conditions and rated and partial shaft horsepowers (shp) are to be calculated as follow.

a. Read theoretical steam rate in [illegible] Theoretical Steam Rate Tables, 1938 Edition, published by The American Society of Mechanical Engineers (Section 4707of General Electric Company Apparatus Handbook).
   1. Convert to lb per hp-hr by multiplying by 0.7455.
b. Read basic efficiency from Table A1 or A2 at rated horsepower and initial press.
c. Read initial temperature correction factor from Table B.
d. Obtain load correction factor from Table C1 or C2 at speed function = $\sqrt[3]{\text{power function}}$.
e. Correct basic efficiency (b.) X (c.) X (d.).
f. Obtain astern rotation loss factor from Table D.
g. Calculate method-exhaust flow:

   $$\text{M.E.F.} = \frac{\text{Power X T.S.R.}}{\text{(e.) X (f.) X 0.98}}$$
h. Determine minimum annulus:

   (g.) ÷ 6000 Back Pressure Inches Hg. A.
i. Determine maximum annulus:

   (g.) ÷ 4000 Back Pressure Inches Hg. A.
j. Select annulus to be used. (See Note 4, Table E.)
k. Determine flow factor:

   (g.) ÷ [BP X (j.)]
l. Obtain excess exhaust loss from Table E.
m. Calculate exhaust-loss factor:

   $$(\text{m.}) = 1.00 - \left(\frac{1}{100}\right)\left(\frac{(\text{l.})}{26.5}\right)\left(\frac{(\text{a.})}{(\text{e.})}\right)$$
n. Calculate engine efficiency:

   (n.) = (e.) X (f.) X (m.)
o. Calculate nonextracting steam rate:

   (o.) = (a.) ÷ (n.)

Example:

See page 52 for example calculations.

### Tolerance on Shipboard Performance Tests

Steam rate guarantees on all turbines are made on the basis of compliance being demonstrated by precision test as specified in the ASME Turbine Test Code PTC6-1964. The instrumentation required by this code cannot be obtained aboard ship. Therefore, when compliance with guarantees is demonstrated by test aboard ship a tolerance for additional testing error is required:

1. If the ship has a torsionmeter and calibrated shaft installed, both approved by the Medium Steam Turbine Generator and Gear Dept. of the General Electric Co., and has properly calibrated throttle pressure, throttle temperature and condenser vacuum gages, and calibrated condensate flow meters, the main propulsion turbine shall meet the guaranteed steam rate after corrections for deviations in throttle pressure, throttle temperature, condenser vacuum and propeller RPM with a 1 percent tolerance for shipboard testing accuracy.
2. If the ship has an approved torsionmeter installed but a standard shaft modulus has been specified in lieu of a calibrated shaft, and has properly calibrated throttle pressure, throttle temperature, and condenser vacuum gages, and calibrated condensate flow meters, the main turbine shall meet the guaranteed steam rate after corrections for deviations in throttle pressure, throttle temperature, condenser vacuum and propeller rpm with a 2% tolerance for shipboard testing accuracy. In either case the test should endure for one hour with steady-state readings on the prime variables.

### Allowable Tolerance on Operating Variables

#### Initial Steam Pressure

For any given load, the steam pressure shall not average more than that specified. In maintaining this average, the pressure shall not exceed 110 percent of that specified. During abnormal conditions, the pressure may swing momentarily to 120 percent of that specified, but the aggregate of such swings shall not exceed 1 percent of the total specified operating life. (3.4.5.1.2.1 MIL-T-17600B.)

#### Initial Steam Temperature

For any given load, the steam temperature shall not average more than that specified. In maintaining this average, the temperature shall not exceed the specified temperature plus 15 F, except that during abnormal conditions the temperature shall not exceed, (a) the specified temperature plus 25 F for not more than 5 percent of the total specified operating life, or (b) the specified temperature plus 50 F for swings of 15 minutes duration or less, aggregating not more than 1 percent of the total specified operating life. (3.4.5.1.2.2 MIL-T-17600B.)

#### Back Pressure

The back pressure at the turbine exhaust flange may vary from a minimum of 0 inches Hg to a maximum of 1½ inches Hg greater than the specified back pressure.

#### Valves Wide Open Throttle Flow

At specified steam conditions, the turbine is designed to pass a valves wide open throttle flow 5 percent greater than that required to develop the maximum specified power and provide the specified extraction flows.

#### Extraction Flow

Extraction openings will be sized to pass the specified extraction flows with nominal pressure drops. Extraction flows of up to 15 percent of the throttle flow may be extracted from a given opening with increased pressure drop.

# Steam Rates

## Single-Cylinder or Cross-Compound

### Speed

The propeller speed for a given horsepower may vary ±3 percent from the specified speed power curve for the ship. During abnormal conditions, the speed for a given power may be decreased by 30 percent provided that the maximum specified $\frac{SHP}{RPM}$ is not exceeded and provided that such operation does not aggregate more than 2 percent of the total specified operating life.

### NUMERICAL EXAMPLE OF METHOD

20,000 Shaft hp - 575 Psig - 840 FTT - 1.75 Inches Hg. Abs.

| Entity | Units | Source | Example Calculation | | | |
|---|---|---|---|---|---|---|
| 1. Power | Shaft hp | Given | 6860 | 10240 | 16000 | 20000 |
| 2. Power Fraction | — | Given Shaft hp / Max. Shaft hp | 0.343 | 0 512 | 0.80 | 1 00 |
| 3. Speed Fraction | — | $\sqrt[3]{(2)}$ | .70 | .80 | .928 | 1.00 |
| 4. Initial Pressure | PSIG | Given | 625 | 625 | 600 | 575 |
| 5. Initial Temperature | °FTT | Given | 860 | 860 | 850 | 840 |
| 6. Back Pressure | Inches Hg. Abs. | Given | 1.00 | 1.25 | 1.50 | 1.75 |
| 7. Theoretical Steam Rate | Lb. hp-hr | Lbs per Kwhr (from T.S.R Table) ×0.7455 | 4 615 | 4.700 | 4 830 | 4.947 |
| 8. Base Efficiency | | Table A @ Max. Shaft hp | 8248 | 8248 | .8260 | .8268 |
| 9. Temperature-correction Factor | - | Table B | 1.001 | 1.001 | 1.000 | 999 |
| 10. Load-correction Factor | -- | Table C | 8540 | .9230 | .9854 | 1.00 |
| 11. Corrected Base Efficiency | | (8), (9), (10) | .7051 | .7620 | .8139 | .8260 |
| 12. Factor for Astern Rotation Loss | | Table D | .997 | .996 | .995 | .994 |
| 13. Method Exhaust Flow (M.E.F.) | Lb/hr | $\frac{(1), (7)}{(11), (12), .98}$ | 46000 | 64700 | 97400 | 123000 |
| 14. Minimum Annulus | ft² | (13) ÷ 6000 (6) | — | | | 11.71 |
| 15. Maximum Annulus | ft² | (13) ÷ 4000 (6) | | | . | 17.57 |
| 16. Annulus Used | ft² | Note 4, Table E | - | | | 13.65 |
| 17. $\frac{F}{\text{Inches Hg. } A_n}$ | Lb/hr/Inches Hg/ft² | $\frac{(13)}{(6), (16)}$ | 3370 | 3790 | 4760 | 5150 |
| 18. Excess Exhaust Loss | BTU, lb | Table E | 5.78 | 7.30 | 11.5 | 13.6 |
| 19. Exhaust-loss Factor | —— | $1.00 - \left(\frac{1}{100}\right)\left(\frac{(18)}{26.5}\right)\left(\frac{[illegible]}{(11)}\right)$ | .9857 | .9830 | .9743 | .9693 |
| 20. Engine Efficiency | —— | (11), (12), (19) [illegible] | .6929 | .7460 | .7890 | .7958 |
| 21. Locus Best Point Non-extracting Steam Rate | Lb/hp-hr | (7) ÷ (20) | 6.66 | 6.30 | 6.12 | 6 22 |

# Steam Rates

## Single-Cylinder or Cross-Compound

### TABLE A1

**Basic Efficiency for Single-cylinder Main Propulsion Turbines (Interpolate for Intermediate Pressures and Ratings)**

| Max SHP | Steam Pressure at Turbine Inlet, Psig 200 | 300 | 400 | 500 | 600 | 700 | 800 | 900 | 1000 |
|---|---|---|---|---|---|---|---|---|---|
| 2000 | 0.737 | 0.726 | 0.718 | 0.710 | 0.702 | 0.694 | 0.687 | 0.682 | 0.675 |
| 2500 | .758 | .749 | .741 | .733 | .726 | .719 | .712 | 706 | 700 |
| 3000 | .774 | .766 | 757 | 750 | .742 | [illegible] | [illegible] | 724 | 717 |
| 3500 | .783 | .775 | .767 | .760 | [illegible] | .746 | [illegible] | 734 | 728 |
| 4000 | .791 | .784 | 776 | [illegible] | [illegible] | 757 | .751 | 745 | .739 |
| 4500 | .797 | .790 | [illegible] | [illegible] | [illegible] | 765 | [illegible] | .754 | .748 |
| 5000 | .801 | 794 | .787 | 781 | .775 | [illegible] | 763 | [illegible] | 752 |
| 6000 | .805 | .799 | .793 | 787 | .782 | [illegible] | [illegible] | .766 | 761 |
| 7000 | 810 | .804 | .799 | 794 | .788 | .783 | .778 | .772 | .768 |
| 8000 | .814 | 809 | 804 | .798 | .793 | 788 | .783 | .777 | .773 |
| 9000 | .818 | 813 | .808 | 802 | .797 | .792 | 787 | .782 | .778 |
| 10000 | .822 | .816 | .811 | 806 | 801 | 796 | 792 | .787 | .783 |

### TABLE A2

**Basic Efficiency for Cross-Compound Main Propulsion Turbines (Interpolate for Intermediate Pressures and Ratings)**

| Max SHP | Steam Pressure at Turbine Inlet, Psig 200 | 400 | 500 | 600 | 700 | 800 | 900 | 1000 | 1250 | 1500 |
|---|---|---|---|---|---|---|---|---|---|---|
| 7000 | 0.810 | 0.799 | 0.794 | 0.788 | 0.783 | 0.778 | 0.772 | 0.768 | 0.755 | 0.742 |
| 8000 | .814 | .804 | .798 | 793 | 788 | .783 | .777 | 773 | .761 | 749 |
| 9000 | .818 | .808 | .802 | 797 | 792 | 787 | .782 | 778 | .767 | 754 |
| 10000 | .822 | .811 | .806 | 801 | 796 | .792 | 787 | 783 | 772 | 760 |
| 11000 | 825 | .814 | 809 | 805 | 800 | .796 | 790 | 787 | 776 | .764 |
| 12000 | 828 | 817 | 812 | 808 | 803 | 799 | 794 | .790 | .780 | 761 |
| 14000 | .832 | 822 | .817 | 814 | 809 | 805 | 800 | 797 | 787 | 776 |
| 16000 | .836 | 826 | 822 | 819 | 814 | 810 | .806 | 802 | 792 | 783 |
| 18000 | .840 | 830 | .826 | 822 | 818 | 814 | 810 | 806 | 798 | 788 |
| 20000 | .843 | .833 | .829 | 826 | 821 | 818 | 814 | 810 | 802 | 793 |
| 22000 | | .835 | .831 | .828 | 824 | 821 | 817 | 813 | 805 | .797 |
| 24000 | | .838 | 834 | .831 | 827 | 824 | 820 | .816 | 808 | 800 |
| 26000 | | .840 | 836 | .833 | 829 | 826 | 822 | 819 | 812 | 803 |
| 28000 | | .841 | 838 | 835 | 831 | .828 | 824 | 821 | 814 | 806 |
| 30000 | | .843 | .839 | .836 | 832 | .829 | 826 | .823 | 816 | 808 |
| 35000 | | | | 840 | 836 | .833 | 831 | 827 | 821 | 814 |
| 40000 | | | | 843 | 839 | 836 | .834 | 830 | 825 | 818 |
| 50000 | | | | 846 | .843 | .840 | 838 | 835 | 830 | .824 |
| 60000 | | | | .849 | 846 | .843 | .841 | .839 | 834 | 828 |
| 70000 | .... | | | .851 | .848 | .846 | .844 | .842 | .837 | .831 |

### TABLE B

**Initial Temperature Correction Factors (Interpolate for Intermediate Pressures and Temperatures)**

| Initial Temp °F | Steam Pressure at Turbine Inlet, Psig 200 | 400 | 600 | 900 | 1500 |
|---|---|---|---|---|---|
| 500 | 0.948 | 0.942 | 0.936 | 0.931 | |
| 510 | 950 | 944 | 938 | 933 | |
| 520 | 952 | 946 | 940 | 936 | |
| 530 | 953 | 948 | .943 | 938 | |
| 540 | 956 | .949 | 945 | 941 | |
| 550 | 958 | 951 | 948 | 943 | |
| 560 | .960 | 954 | 950 | .946 | |
| 570 | 962 | 956 | 952 | 948 | |
| 580 | 964 | 958 | 953 | 950 | |
| 590 | 965 | 960 | 955 | 953 | |
| 600 | 967 | 962 | 958 | .954 | 0.949 |
| 610 | .969 | 964 | 960 | 956 | 951 |
| 620 | 970 | .966 | 962 | 958 | 954 |
| 630 | 972 | 967 | 964 | 960 | 956 |
| 640 | .974 | 969 | .966 | 962 | .958 |
| 650 | .975 | .971 | .968 | .964 | 960 |
| 660 | .977 | .973 | 970 | 966 | 962 |
| 670 | 978 | 975 | .971 | 968 | 965 |
| 680 | .980 | .976 | 973 | 970 | .967 |
| 690 | 981 | .978 | 975 | 972 | 969 |
| 700 | 983 | 980 | 977 | 974 | 971 |
| 710 | .984 | 981 | 979 | 976 | 973 |
| 720 | 985 | 982 | 980 | 978 | 975 |
| 730 | .986 | .984 | 982 | 979 | 977 |
| 740 | 988 | 985 | 983 | 981 | 979 |
| 750 | .989 | 987 | 985 | 983 | 981 |
| 760 | 990 | 989 | 987 | 985 | 983 |
| 770 | 991 | 990 | 988 | 987 | 985 |
| 780 | 993 | 991 | 990 | 988 | 987 |
| 790 | .994 | 993 | 991 | 990 | 989 |
| 800 | 995 | .994 | 993 | 992 | 991 |
| 810 | 996 | 995 | 994 | 993 | 993 |
| 820 | 997 | 997 | 996 | 995 | 995 |
| 830 | 998 | 998 | 997 | 997 | 996 |
| 840 | 999 | .999 | 999 | 998 | .998 |
| 850 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 |
| 860 | 1.001 | 1.001 | 1.001 | 1.001 | 1.002 |
| 870 | 1.002 | 1.003 | 1.002 | 1.002 | 1.003 |
| 880 | 1.003 | 1.004 | 1.003 | 1.004 | 1.005 |
| 890 | 1.003 | 1.005 | 1.005 | 1.005 | 1.006 |
| 900 | 1.004 | 1.006 | 1.006 | 1.006 | 1.007 |
| 910 | | 1.006 | 1.007 | 1.008 | 1.008 |
| 920 | | 1.007 | 1.008 | 1.009 | 1.010 |
| 930 | | 1.008 | 1.009 | 1.010 | 1.011 |
| 940 | | 1.009 | 1.010 | 1.011 | 1.012 |
| 950 | | 1.010 | 1.011 | 1.012 | 1.014 |
| 1000 | | | 1.015 | 1.017 | 1.020 |
| 1050 | | | 1.018 | 1.021 | 1.026 |

*New information.*

*Data subject to change without notice*

# Steam Rates

## Single-Cylinder or Cross-Compound

### TABLE C1

**Load Correction Factors**
**(Interpolate for Intermediate Speeds) (Fraction of Speed = $\sqrt[3]{\text{Fraction Power}}$)**

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Fraction of Max Specified SPEED | | 0.60 | 0.65 | 0.70 | 0.75 | 0.80 | 0.85 | 0.90 | 0.92 | 0.94 | 0.96 | 0.98 | 1.00 |
| Fraction of Max Specified POWER | | .216 | .275 | .343 | .432 | .512 | .614 | .729 | .779 | .831 | .885 | .941 | 1.00 |
| Efficiency Factor | Single Flow | .774 | 815 | .854 | .890 | .923 | .951 | .975 | .983 | .989 | .994 | .998 | 1.00 |
| | Double Flow | .766 | [illegible] | [illegible] | [illegible] | [illegible] | [illegible] | 965 | .973 | .979 | .984 | .988 | .990 |

1. [illegible]is table is valid for 10 [illegible] efficiency factors are based on the locus of best valve points. [illegible]gree change in initial tem[illegible]

### TURBINES WITH IMPROVED LIGHT-LOAD PERFORMANCE

The basic method determines performance for straight thru turbines designed for endurance operation at 75 to 100 percent power. Performance at the lower powers and speeds can be significantly improved with some penalty in performance at the higher powers and speeds by increasing the turbine wheel speed and by adding a cruising section. The cruising section is inactivated at the higher powers by bypassing or by paralleling. Such turbines are designed specifically for a given application and performance will vary, depending on the evaluations at the specified points of operation. Guaranteed performance on such designs can be obtained by factory inquiry. However, in order to indicate the degree of tilting of performance possible the following load factors substituted for Table C of the method will give reasonable estimating performance for designs with cruising points at less than 25 percent load.

### TABLE C2

**Load Correction Factors**
**(I[illegible]polate for Intermediate Speeds)**

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Fraction of Max Specified SPEED | | 0.45 | 0.50 | 0.55 | 0.60 | 0.65 | 0.70 | 0.75 | 0.80 | 0.85 | 0.90 | 0.95 | 1.00 |
| Fraction of Max Specified POWER | | .091 | .125 | .166 | .216 | .275 | .343 | 422 | .512 | .614 | 729 | .857 | 1.00 |
| Efficiency Factor | 2 row/1 row open close valve gear | .683 | .747 | .801 | .843 | 876 | .903 | .924 | .941 | .956 | .967 | .976 | .985 |
| | Series Parallel | .737 | .795 | .842 | .877 | 888 | 895 | 904 | 914 | .926 | .941 | .956 | .973 |
| | Interstage Bypass | .737 | .795 | .842 | 877 | .898 | .909 | .916 | .920 | .927 | .935 | .946 | .958 |

### TABLE D

**Factors for Astern Rotation Loss**

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Back Pressure, Inches Hg | 0.5 | 1.0 | 1.5 | 2.0 | 2.5 | 3.0 | 4.0 | 5.0 |
| Factor | .998 | .997 | .995 | .993 | .992 | .990 | .987 | .983 |

# Steam Rates

## Single-Cylinder or Cross-Compound

### TABLE E

**Excess Exhaust Loss (E.E.L.), BTU/Lb. (Interpolate for Intermediate Exhaust Flows and Speeds)**

#17 p52

| Exhaust Flow, Lb/Hr ÷ Exh Press, Inches Hg X Annulus Area, ft² | Percent Speed 40 | 50 | 60 | 70 | 80 | 90 | 100 |
|---|---|---|---|---|---|---|---|
| | Excess Exhaust Loss, BTU/Lb | | | | | | |
| 8000 | | | | | 26.8 | 25.8 | 25.8 |
| 7500 | | | | 26.6 | 24.6 | 23.6 | 23.6 |
| 7000 | | | | 24.2 | 22.3 | 21.5 | 21.5 |
| 6500 | | | 24.3 | 21.7 | 19.9 | 19.1 | 19.2 |
| 6000 | | | 21.9 | 19.3 | 17.6 | 16.8 | 17.2 |
| 5500 | | | 19.3 | 16.7 | 15.1 | 14.5 | 15.0 |
| 5000 | | 19.9 | 16.5 | 14.1 | 12.7 | 12.2 | 13.0 |
| 4500 | | 16.8 | 13.6 | 11.3 | 10.3 | 10.2 | 11.3 |
| 4000 | 17.6 | 13.7 | 10.7 | 8.8 | 8.1 | 8.4 | 9.8 |
| 3500 | 14.0 | 10.4 | 7.8 | 6.3 | 6.2 | 7.0 | 6.8 |
| 3000 | 10.3 | 7.0 | 5.1 | 4.3 | 4.7 | 6.1 | 8.3 |
| 2500 | 6.1 | 3.8 | 2.7 | 2.8 | 3.9 | 6.0 | 9.2 |
| 2000 | 2.3 | 1.2 | 1.1 | 2.2 | 4.4 | 7.6 | 11.8 |
| 1500 | -1.0 | -0.5 | 1.0 | 3.6 | 7.3 | 11.8 | 17.4 |
| 1300 | -1.9 | -0.7 | 1.7 | 4.9 | 9.3 | 14.3 | 20.6 |
| 1100 | -2.4 | -0.2 | 3.1 | 7.0 | 12.1 | 17.9 | |
| 900 | 2.4 | 0.8 | 5.0 | 9.7 | 15.8 | | |

1. Excess exhaust loss is equal to used energy end point (U.E.E.P.) minus the expansion line end point (E.L.E.P.) A negative value of E.E.L. means that the internal efficiency is better than that indicated by the E.L.E.P.
2. Method Exhaust Flow =
   SHP X T.S.R. (lb/hp-hr)
   (Table A X Table B X Table C X Table D X 0.980
3. E.L. Factor =
   $$1.00 - \frac{1}{100} \times \frac{\text{E.E.L.}}{26.5} \times \frac{\text{T.S.R. (lb/hp-hr)}}{\text{(Table A X Table B X Table C)}}$$
4. Available Annulus Areas (ft )

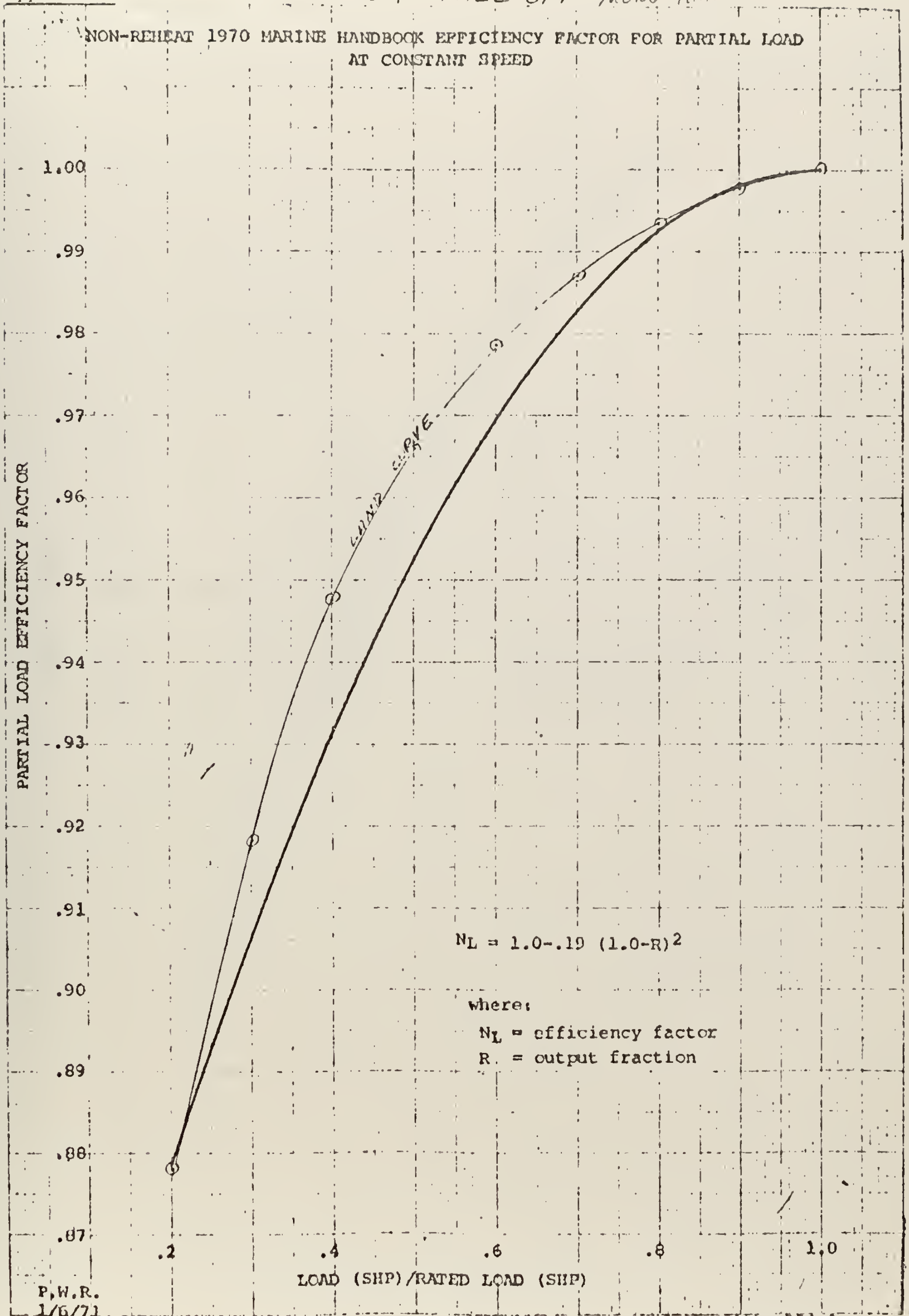
| | | | |
|---|---|---|---|
| Single-casing, Single-flow | 2.40 | 4.99 | 6.76 |
| Cross-compound, Single-flow | 9.17 | 13.65 | 18.20 |
| Cross-compound, Double-flow | 24.96 | 30.0 | |

5. For good practice $\frac{\text{Exhaust Flow, lb/Hr. @ 100\% Speed}}{\text{Exh. Press., inches Hg X Annulus Area, ft}^2}$ should not be greater than 6000 nor less than 4000.
6. However those cases where $\frac{\text{Exhaust Flow, lb/Hr. @ 100\% Speed}}{\text{Annulus Area}}$ is greater than 10,000 lb/hr/ft² should be referred to the fctory for investigation of mechanical strength.

FOR VARIABLE PRESSURE - CONST. AREA

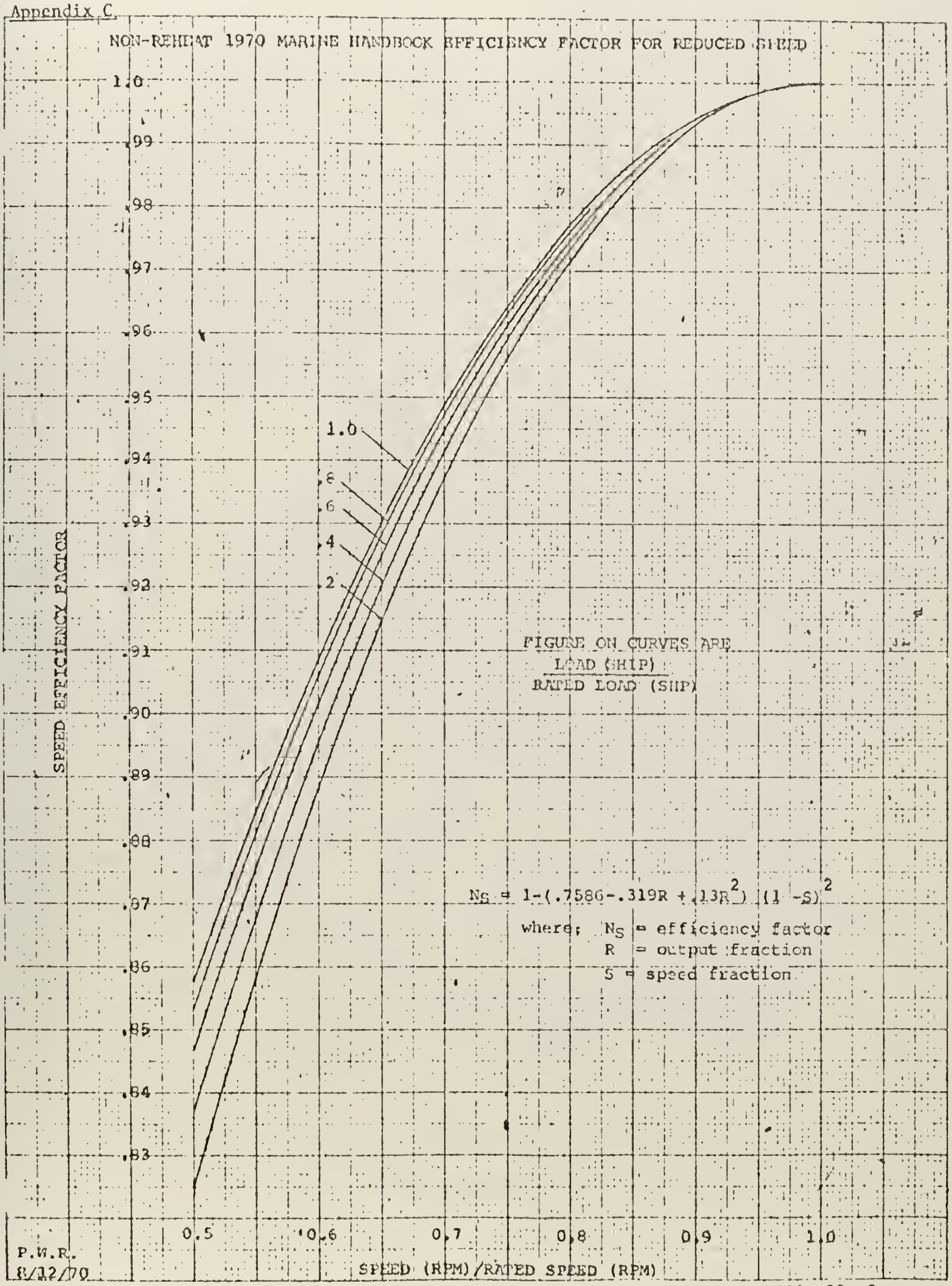
THIS WILL NOT FALL OFF MORE THAN THIS





FOR VARIABLE PRESSURE BACK THIS FACTOR OUT





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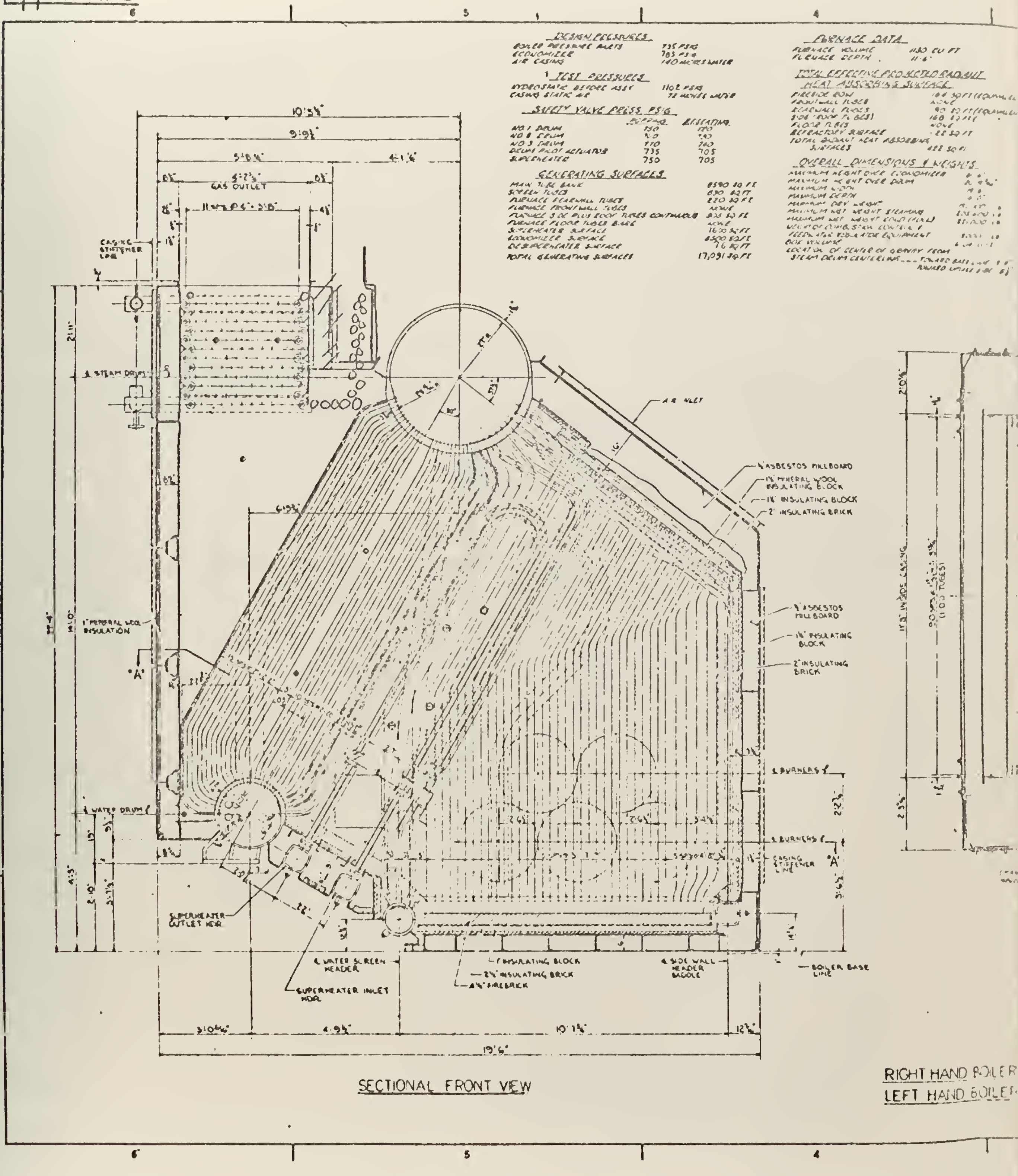
Appendix D

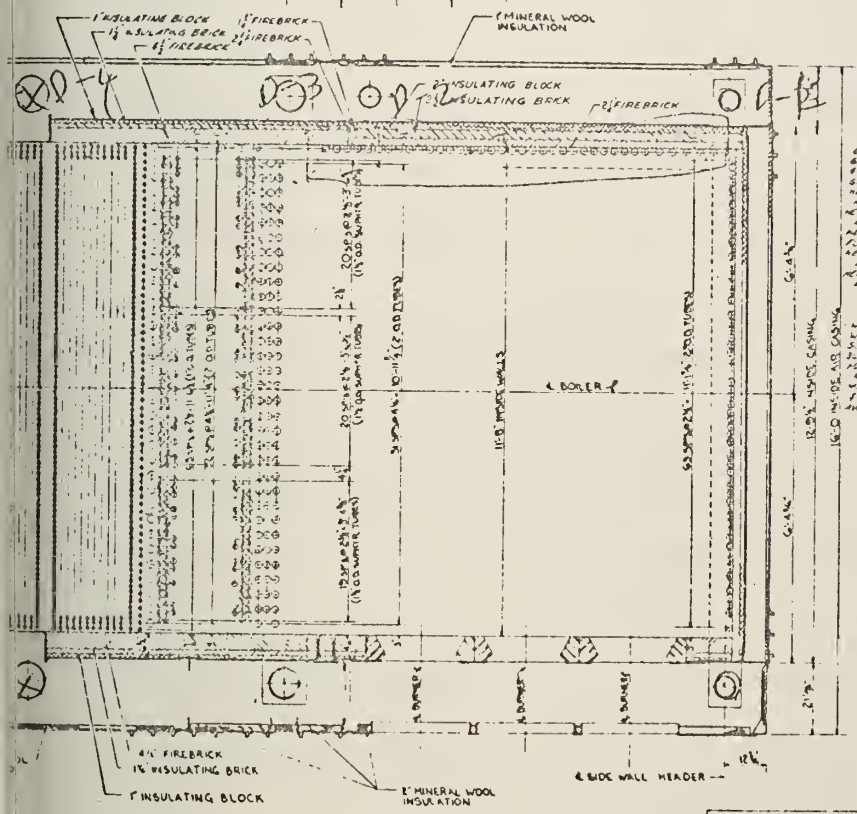

DESIGN PRESSURES
TEST PRESSURES
SAFETY VALVE PRESS. PSIG
GENERATING SURFACES
TOTAL GENERATING SURFACES 17,091 SQ FT
FURNACE DATA
TOTAL EFFECTIVE PROJECTED RADIANT HEAT ABSORBING SURFACE
OVERALL DIMENSIONS & WEIGHTS
GAS OUTLET
CASING STIFFENER LINE
STEAM DRUM
WATER DRUM
AIR INLET
1/2" ASBESTOS MILLBOARD
1 1/2" MINERAL WOOL INSULATING BLOCK
1 1/2" INSULATING BLOCK
2" INSULATING BRICK
BURNERS
SUPERHEATER OUTLET HDR.
WATER SCREEN HEADER
SUPERHEATER INLET HDR.
INSULATING BLOCK
2 1/2" INSULATING BRICK
4 1/2" FIREBRICK
SIDE WALL HEADER
BOILER BASE LINE
10'5 3/4"
9'9 1/2"
19'6"
SECTIONAL FRONT VIEW
RIGHT HAND BOILER
LEFT HAND BOILER

| REV | DESCRIPTION |
|---|---|
| B | DESIGN PRESS WAS 700 PSI HYDROSTATIC PRESS WAS 1050 PSI |
| C | CERTIFIED [illegible] CHANGE: GENERATING SURFACE [illegible] SQ FT, [illegible] SQ FT, TOTAL HAS [illegible] SQ FT, TUBE DATA: DESUPERHEATER WAS 2" OD, .170" MWT, NO. 1 |

## ANTICIPATED PERFORMANCE

| | 10 PCT | ENDURANCE | BEACH (SPEED) | F P | MAXIMUM OVERLOAD |
|---|---|---|---|---|---|
| RATE OF OPERATION, PERCENT | 10 | 43.6 | 68.7 | 100 | 120 |
| TOTAL STEAM GENERATED, LB/HR | 20100 | 95700 | 150000 | 220000 | 264000 |
| SUPERHEATED STEAM, LB/HR | 20100 | 86700 | 108000 | 195900 | 235900 |
| SATURATED OR DESUPERHEATED STEAM, LB/HR | — | 9000 | 35000 | 3100 | 6000 |
| BOILER DRUM PRESSURE, PSIG | 600 | 615 | 625 | 650 | 685 |
| SUPERHEATER OUTLET PRESS, PSIG | 600 | 600 | 600 | 600 | 600 |
| SUPERHEATER OUTLET TEMP, °F | 785 | 850 | 855 | 860 | 865 |
| DESUPERHEATER STEAM OUT PRESS, PSIG | — | 595 | 560 | 595 | 599 |
| DESUPERHEATER OUT TEMP °F | — | 571 | 640 | 550 | 557 |
| ECON INLET PRESSURE PSIG | 625 | 615 | 645 | 680 | 735 |
| ECON INLET TEMPERATURE, °F | 246 | 246 | 246 | 246 | 246 |
| ECON OUTLET PRESSURE PSIG | 600 | 615 | 625 | 650 | 685 |
| ECON OUTLET TEMPERATURE, °F | 350 | 356 | 336 | 347 | 359 |
| CASING AIR INLET TEMP °F | 100 | 100 | 100 | 100 | 100 |
| TOTAL AIR FLOW LB/HR | 35400 | 126000 | 163800 | 247500 | 328500 |
| FURNACE AIR INLET TEMP °F | 100 | 100 | 100 | 100 | 100 |
| TOTAL OIL FLOW, LB/HR | 550 | 7230 | 10180 | 15860 | 18550 |
| ANTICIPATED EFFICIENCY, PCT | 82 | 85 | 87.9 | 87 | 85.9 |
| GUARANTEED EFFICIENCY, PCT | — | 86 | — | 87 | — |
| RADIATION & UNACCOUNTED FOR LOSSES, PCT | 2.5 | 1.5 | 1.2 | 0.8 | 0.8 |
| EXCESS AIR PCT | 60 | 10.5 | 15 | 15 | 12.5 |
| PERCENT CARBON DIOXIDE | 9.85 | 13.5 | 13.9 | 13.9 | 13.5 |
| NO OF BURNERS IN OPERATION | 5 | 5 | 5 | 5 | 5 |
| THROTTLE OR NONTHROTTLE OF AIR DOORS | | NONTHROTTLE | | * | * |
| DRAFT LOSS - TOTAL INCHES WATER | .8 | 9.0 | 15.0 | 35.8 / 46 | 51 / 64.9 |
| THROUGH DOUBLE CASING | .1 | 1.4 | 2.4 | 4.6 / 6.0 | 2.1 / 8.7 |
| THROUGH BURNER REGISTER | .6 | 5.8 | 8.6 | 20.2 / 26.4 | 28.0 / 35.7 |
| THROUGH BOILER & SUPERHEATER | — | .9 | 1.3 | 3.6 / 4.9 | 5.1 / 6.8 |
| THROUGH ECONOMIZER | .1 | 1.7 | 2.9 | 2.6 / 9.7 | 11.8 / 14.3 |
| ANTICIPATED EFFICIENCY PCT BASED ON 840 CU FT AIR/LB OF OIL | — | 87.9 | 87.6 | 86.6 | |

* DRAFT LOSSES BASED ON 840 CU FT OF 100°F AIR/LB OF OIL.




PLAN SECTION 'A-A'

B SHOWN
12A OPPOSITE HD.

## ANTICIPATED PERFORMANCE (CONT)

| | 10 PCT | ENDURANCE | BEACH SPEED | F P | MAXIMUM OVERLOAD |
|---|---|---|---|---|---|
| GAS TEMP LEAVING SUPHTR SCREEN °F | 1500 | 2265 | 2460 | 2555 | 2575 |
| GAS TEMP LEAVING SUPHTR °F | 1110 | 675 | 795 | 1445 | 1585 |
| GAS TEMP LEAVING MAIN BANK °F | 495 | 553 | 680 | 700 | 745 |
| GAS TEMP LEAVING ECON °F | 263 | 3 3 | 335 | 385 | 420 |
| HEAT REL KBTU/HR/SQ FT RAD HEAT ABSORBING SURFACE | 63.6 | 300.8 | 428 | 632.5 | 778 |
| HEAT REL KBTU/HR/SQ FT TOTAL HEAT ABS SURFACE | 4.7 | 27.6 | 10.94 | 16.3 | 19.91 |
| HEAT REL KBTU/HR/CU FT FURN VOL | 25.9 | 48.2 | 167 | 251 | 306 |
| FURN HEAT ABSORPTION KB/HR/SQ FT | 27.1 | 98.3 | 127 | 170 | 185.5 |
| HEAT ABSORPTION FIRST SCREEN ROW KB/HR/SQ FT | 16.4 | 66.5 | 85.5 | 150 | 292 |
| HEAT ABSORPTION MAXIMUM (FURN) KB/HR/SQ FT | 27.1 | 98.5 | 127 | 170 | 189.6 |

### ANTICIPATED METAL TEMPERATURE F°

| | 10 PCT | ENDURANCE | BEACH SPEED | F P | MAXIMUM OVERLOAD |
|---|---|---|---|---|---|
| WATER SCREEN TUBES OUTSIDE | 550 | 560 | 560 | 6 5 | 625 |
| WATER SCREEN TUBES INSIDE | 540 | 540 | 545 | 450 | 555 |
| SUPHTR TUBES OUTSIDE MAX | 850 | 925 | 950 | 935 | 935 |
| SUPHTR TUBES INSIDE MAX | 825 | 915 | 936 | 900 | 900 |
| MAX INNER CASING TEMP AT 6TH ROW OF MAIN BANK | 530 | 550 | 575 | 600 | 625 |
| MAX OUTER CASING TEMP AT WAY OF STRUCTURAL TIES | 350 | 350 | 350 | 350 | 350 |
| AVERAGE OUTER CASING TEMP | 140 | 140 | 140 | 140 | 160 |

### TUBE DATA

| LOCATION | O.D. | MWT | NO. |
|---|---|---|---|
| SIDEWALL & ROOF | 2" | .134" | 64 |
| REARWALL | 2" | .134" | 41 |
| SCREEN BANK | 2" | .134" | 95 |
| SUPERHEATER | 1¼" | .120" | 162 |
| MAIN BANK | 1" | .095" | 1432 |
| MAIN BANK | 2" | .150" | 33 |
| ECONOMIZER | 1" | .135" | 140 |
| DESUPERHEATER | 2" | .180" | 4 |

### REFERENCE DRAWINGS

| TITLE | C.E. INC. DWG. NO. |
|---|---|
| ARRANGEMENT OF EXTERNAL CONNECTIONS & FITTINGS | F-167-430 |
| ASSEMBLY OF FLOOR CASING & BOILER SEATING | F-167-433 |

FAST COMBAT SUPPORT SHIP
AOE-3
MAIN BOILERS
GENERAL ARRANGEMENT
SECTIONAL FRONT & PLAN VIEW

COMBUSTION ENGINEERING INC.
WINDSOR, CONN.

DWG NO F-167-431-C

PLATE F-1

Appendix D

## SUMMARY OF BOILER CALCULATIONS

| | Cruise Power | | Full Power | |
|---|---|---|---|---|
| | Temp. (°F) | Pressure (psig) | Temp. (°F) | Pressure (psig) |
| **Waterside:** | | | | |
| Economizer Inlet | 280 | 415 | 280 | 1275 |
| Economizer Outlet | 448 | 415 | 407 | 1275 |
| Drum | 448 | 415 | 578 | 1275 |
| Superheater Outlet | 758 | 400 | 997 | 1200 |
| Desuperheater Outlet | - - | - - | 655 | 1160 |
| **Fireside:** | | | | |
| Firebox | 950 | | 2752 | |
| After Superheater | 816 | | 1757 | |
| After Main Bank | 492 | | 761 | |
| Stack | 364 | | 355 | |
| **Flow:** | | | | |
| Steam, Superheated | 25,000 Lb/Hr | | 230,000 Lb/Hr | |
| Steam, Desuperheated | 0 Lb/Hr | | 30,000 Lb/Hr | |
| Air | 194,000 Lb/Hr | | 293,238 Lb/Hr | |
| **Draft Losses:** | | | | |
| Boiler (less burners) | 4.36 in. water | | 20 in. water | |

Appendix D

## CRUISE MODE BOILER PERFORMANCE OF THE SCREEN BANK

April 5, 1972

| | | |
|---|---|---|
| 1. | Tube Size & Thickness | 2 |
| 2. | No Rows Deep | 3 |
| 3. | No Tubes Wide | 32-31-32 |
| 4. | Rating | |
| 5. | Lb. Gas/hr W | 194,000 |
| 6. | Free Area | 75.8 |
| 7. | Mass Flow | 2559 |
| 8. | Eff Surface S | 595 |
| 9. | Gas Temp ent $T_1$ | 950 |
| 10. | Est Gas Temp lvg | 900 |
| 11. | Avg Gas Temp | 925 |
| 12. | Cp | .268 |
| 13. | Sat Steam Temp Ts | 448 |
| 14. | Beam Length | 4.05 |
| 15. | Rr | .94 |
| 16. | Diam Correction | .812 |
| 17. | Fa | .79 |
| 18. | Rc (Curve M- ) | 9.75 |
| 19. | Rc Corrected | 6.249 |
| 20. | R | 7.189 |
| 21. | Aloge RS/WCp | 1.086 |
| 22. | $T_1$ - Ts | 502 |
| 23. | $T_2$ - Ts | 462 |
| 24. | Gas Temp lvg $T_2$ | 910 |
| 25. | Gas Temp ent Shtr | |
| 26. | Btu/lb Gas ent Shtr | |
| 27. | Mm Btu/hr Gas ent Shtr | |
| 28. | Mm Btu/hr Absorbed as SM | |
| 29. | Mm Btu/hr Gas lvg Shtr | |
| 30. | Btu/lb Gas lvg Shtr | |
| 31. | Gas Temp lvg Shtr | |
| 32. | Draft Loss/Row | |
| 33. | f | .0248 |
| 34. | Draft Loss | |

| | |
|---|---|
| Long Spacing | 2 1/2 |
| Trans Spacing | 4 1/4 |
| In Line | (✓) |
| Stggd | ( ) |
| Product | 10.62 |

form 231

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Appendix D

## CRUISE MODE BOILER PERFORMANCE OF THE SUPERHEATER

April 5, 1972

| | | | |
|---|---|---|---|
| 1. | Tube Size and Thickness | 1 1/2 | |
| 2. | No Tubes Wide and Spacing | 62@2 1/8 | |
| 3. | No Tubes Deep | 6 | |
| 4. | Effective Length/Tube | | |
| 5. | Effective Heating Surface S | 1658 | |
| 6. | Gas Free Area | 46.1 | |
| 7. | Rating | | |
| 8. | Gas - Flow Wg | 194,000 | |
| 9. | Temp ent $T_1$ | 910 | |
| 10. | Temp lvg $T_2$ | 835 | 816 by formulas |
| 11. | Temp avg | 872 | |
| 12. | Specific Heat Cp | .267 | |
| 13. | Steam - Flow Ws | 25,000 | |
| 14. | Temp ent $t_1$ | 448 | |
| 15. | Temp lvg $t_2$ | 696 | 758 by formulas |
| 16. | Temp Rise | 248 | |
| 17. | Enthalpy Change | 155 | |
| 18. | Specific Heat Cs | .625 | |
| 19. | Gas Mass Flow G | 4208 | |
| 20. | Transfer Rate R | 15 | |
| 21. | RS/ WgCp | | |
| 22. | WgCp/ WsCs | 3.31 | |
| 23. | $(T_1 - T_2) / (T_1 - t_1)$ | | |
| 24. | $T_1 - t_1$ | | |
| 25. | $T_1 - T_2$ | | |
| 26. | $T_2$ | | |
| 27. | $WgCp(T_1 - T_2) / Ws$ | | |
| 28. | Btu/lb Stm ent | | |
| 29. | Btu/lb Stm lbg | | |
| 30. | Shtd Stm Temp | | |
| 31. | No Steam Passes | | |
| 32. | Avg Specific Volume v | | |
| 33. | Actual Element Length | | |
| 34. | Equiv Length Bends, etc. | | |
| 35. | Total Equiv Length L | | |
| 36. | Lb Shtd Stm/min/Element F | | |
| 37. | Tube Inside Diameter D | | |
| 38. | $F^{1.85}$ | | |
| 39. | $D^{4.97}$ | | |
| 40. | Shtr. Pressure Drop | | |

form 324

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Appendix D

## CRUISE MODE BOILER PERFORMANCE OF THE MAIN BANK

April 5, 1972

| | | | | |
|---|---|---|---|---|
| 1. | Tube Size & Thickness | 1 | Long Spacing | 2 |
| 2. | No Rows Deep | 28 | Trans Spacing | 1 17/32 |
| 3. | No Tubes Wide | 91 | In Line | (✓) |
| | | | Stggd | ( ) |
| 4. | Rating | | Product | 3.0625 |
| 5. | Lb. Gas/hr W | 194,000 | | |
| 6. | Free Area | 48.7 | | |
| 7. | Mass Flow | 3984 | | |
| 8. | Eff Surface S | 7812 | | |
| 9. | Gas Temp ent $T_1$ | 816 | | |
| 10. | Est Gas Temp lvg | 475 | | |
| 11. | Avg Gas Temp | 646 | | |
| 12. | Cp | .261 | | |
| 13. | Sal Steam Temp Ts | 448 | | |
| 14. | Beam Length | 2.45 | | |
| 15. | Rr | .46 | | |
| 16. | Diam Correction | 1.00 | | |
| 17. | Fa | 1.022 | | |
| 18. | Rc (Curve M-49B) | 13.0 | | |
| 19. | Rc Corrected | 13.286 | | |
| 20. | R | 13.746 | | |
| 21. | Aloge RS/WCp | 8.34 | | |
| 22. | $T_1$ - Ts | 368 | | |
| 23. | $T_2$ - Ts | 44 | | |
| 24. | Gas Temp lvg $T_2$ | 492 | | |
| | RN = 5100 | | | |
| 25. | Gas Temp ent Shtr | | | |
| 26. | Btu/lb Gas ent Shtr | | | |
| 27. | Mm Btu/hr Gas ent Shtr | | | |
| 28. | Mm Btu/hr Absorbed as SM | | | |
| 29. | Mm Btu/hr Gas lvg Shtr | | | |
| 30. | Btu/lb Gas lvg Shtr | | | |
| 31. | Gas Temp lvg Shtr | | | |
| 32. | Draft Loss/Row | .31 | | |
| 33. | f | .038 | | |
| 34. | Draft Loss 38 Tubes | .42" | | |

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## Appendix D

### CRUISE MODE PRELIMINARY HEAT BALANCE OF THE BOILER

**Steam Side**

Input Conditions: Water at 280°F = 249 Btu/Lb

Output Conditions Steam at 400 psig and 758°F = 1394 Btu/Lb

Flow Rate: 25000 Lb/Hr

Heat Gain: (25000) Lb/Hr (1394 - 249) Btu/Lb = 28,625,000 Btu/Hr

**Air Side**

Input Conditions: 950°F = 219 Btu/Lb

Flow Rate: 194,000 Lb/Hr

| | | |
|---|---|---|
| Input Heat: (194,000) Lb/Hr (219) Btu/Lb = | 42,486,000 | Btu/Hr |
| Less Steam Side Heat Gain | -28,625,000 | Btu/Hr |
| Output Heat: | 13,861,000 | |

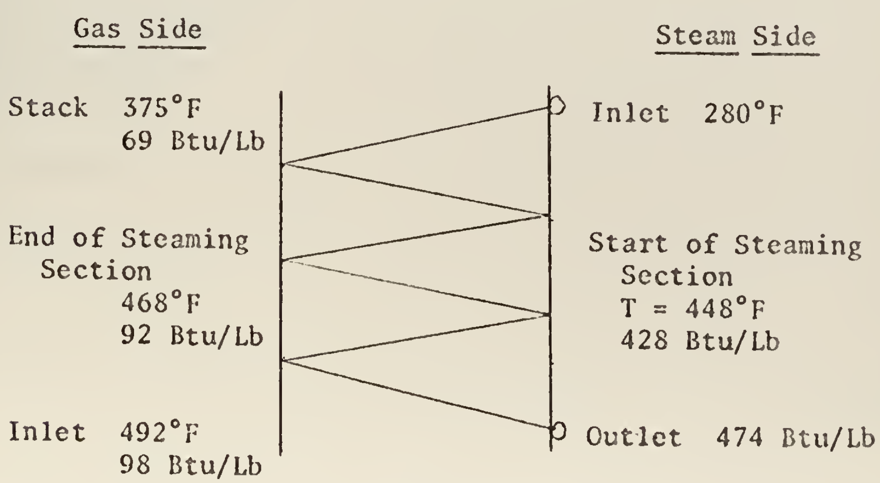
Output Conditions: 71 Btu/Lb and 384°F Stack Temperature

Goal is Stack Temperature of 375°F so Heat Gain to Steam Side

would be: (194,000) Lb/Hr (0.261) Btu/Lb°F (575)°F = 29,114,550 Btu/Hr

---

### CRUISE MODE ECONOMIZER HEAT BALANCE



Appendix D

## CRUISE MODE BOILER PERFORMANCE OF THE

## ECONOMIZER STEAMING SECTION

| | | |
|---|---|---|
| 1. | Tube Size & Thickness | |
| 2. | No Rows Deep | |
| 3. | No Tubes Wide | |
| | S.S. 15 Wide x 11.67' Bts 8 Rows High | |
| 4. | Rating | |
| 5. | Lb. Gas/Hr W | 194,000 |
| 6. | Free Area (15)(11.67)(.134) | 23.5 |
| 7. | Mass Flow | 8255 |
| 8. | Eff Surface S | 4667 |
| 9. | Gas Temp ent $T_1$ | 492 |
| 10. | Est Gas Temp lvg | 468 |
| 11. | Avg Gas Temp | 480 |
| 12. | Cp | .255 |
| 13. | Sat Steam Temp Ts | 448 |
| 14. | Beam Length | |
| 15. | Rr | |
| 16. | Diam Correction | |
| 17. | Fa | |
| 18. | Rc (curve M- ) | |
| 19. | Rc Corrected | |
| 20. | R | 8.9 |
| 21. | Aloge RS/WCp .789 | 2.20 |
| 22. | $T_1$ - Ts | 44 |
| 23. | $T_2$ - Ts | 20 |
| 24. | Gas Temp lvg $T_2$ | 468 |
| 25. | Gas Temp ent Shtr | |
| 26. | Btu/lb Gas ent Shtr | |
| 27. | Mm Btu/hr Gas ent Shtr | |
| 28. | Mm Btu/hr Absorbed as SM | |
| 29. | Mm Btu/hr Gas lvg Shtr | |
| 30. | Btu/lb Gas lvg Shtr | |
| 31. | Gas Temp lbg Shtr | |
| 32. | Draft Loss/Row | |
| 33. | f | |
| 34. | Draft loss | |

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form 231

Appendix D

## CRUISE MODE BOILER PERFORMANCE OF THE MODIFIED ECONOMIZER STEAMING SECTION

| | | |
|---|---|---|
| 1. | Tube Size & Thickness | |
| 2. | No Rows Deep | |
| 3. | No Tubes Wide | |
| | S.S. 17 wide x 13' Bts 7 High | |
| 4. | Rating | |
| 5. | Lb. Gas/hr W | 194,000 |
| 6. | Free Area 17 x 13 x .134 | 29.6 |
| 7. | Mass Flow | 6554 |
| 8. | Eff Surface S | |
| 9. | Gas Temp ent $T_1$ | 492 |
| 10. | Est Gas Temp lvg | 468 |
| 11. | Avg Gas Temp | 480 |
| 12. | Cp | .255 |
| 13. | Sat Steam Temp Ts | 448 |
| 14. | Beam Length | |
| 15. | Rr | |
| 16. | Diam Correction | |
| 17. | Fa | |
| 18. | Rc (Curve M- ) | |
| 19. | Rc Corrected | |
| 20. | R | 7.9 |
| 21. | Aloge RS/WCp | 2.20 |
| 22. | $T_1$ - Ts | 44 |
| 23. | $T_2$ - Ts | 20 |
| 24. | Gas Temp lvg $T_2$ | 468 |
| 25. | Gas Temp ent Shtr | |
| 26. | Btu/lb Gas ent Shtr | |
| 27. | Mm Btu/hr Gas ent Shtr | |
| 28. | Mm Btu/hr Absorbed as SM | |
| 29. | Mm Btu/hr Gas lvg Shtr | |
| 30. | Btu/lb Gas lvg Shtr | |
| 31. | Gas Temp lvg Shtr | |
| 32. | Draft Loss/Row | |
| 33. | f | |
| 34. | Draft Loss | |

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Appendix D

## CRUISE MODE BOILER PERFORMANCE OF THE

## NON-STEAMING ECONOMIZER

| | | Standard Economizer | | Modified Economizer |
|---|---|---|---|---|
| 1. | Tube Size and Thickness | 2" S.S. | | |
| 2. | No Rows High | 24 | | 20 |
| 3. | No Tubes Wide | 15 | | 17 |
| 4. | Length between Tube Sheets | 11.67 | | 13.0' |
| 5. | Heating Surface/Tube | | | |
| 6. | Total Heating Surface S | | calc. by formula 8893 | 10100 |
| 7. | Rating | | | |
| 8. | Lb Gas/hr Wg | 194,000 | | |
| 9. | Lb Water/hr Ww | 25,000 | | |
| 10. | Gas Temp ent $T_1$ | 468 | | |
| 11. | Est Gas Temp lvg | 375 | | |
| 12. | Avg Gas Temp | 422 | | |
| 13. | Water Temp ent $t_1$ | 280 | | |
| 14. | Est Water Temp lvg | 448 | | |
| 15. | Avg Water Temp | 364 | | |
| 16. | Free Area (15)(11.67)(.134) | 23.5 | | |
| 17. | Mass Flow | 8255 | | |
| 18. | Transfer Rate R | 10.1 | | 8.9 |
| 19. | Cp | .254 | | |
| 20. | Cw | 1.053 | | |
| 21. | RS/Wg Cp | | | |
| 22. | WgCp / Ww Cw | 1.871 | | |
| 23. | $(T_1 - T_2) / (T_1 - t_1)$ | | | |
| 24. | $T_1 - t_1$ | | | |
| 25. | $T_1 - T_2$ | | | |
| 26. | $t_2 - t_1$ | | | |
| 27. | Water Temp lvg Econ $t_2$ | | | |
| 28. | Gas Temp lvg Econ $T_2$ | 364 | | |
| 29. | Avg Specific Volume | | | |
| 30. | Lbs Water/Element/hr | | | |
| 31. | Water Velocity | | | |
| 32. | Length/Element | | | |
| 33. | Econ Pressure Drop | | | |
| 34. | Draft Loss/Ten Tubes | | | |
| 35. | Total Draft Loss (In. Water) | 7.56 | | 3.94 |

form 232

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Appendix D

FIRED MODE BOILER PERFORMANCE OF THE FURNACE
(Navy Distillate Fuel Oil Fired)

| | | Full Power | Overload |
|---|---|---|---|
| 1. | Rating % | 100 | 120 |
| 2. | Total lb Steam/hr | 230,000 | 276,000 |
| 3. | Lb Dshtd Steam/hr | 30,000 | 50,000 |
| 4. | Pressure at S.O. | 1200 | 1165 |
| 5. | Total Steam Temp | 950 | 925 |
| 6. | Feedwater Temp | 280 | 280 |
| 7. | Pressure at D.S.O. | 1160 | 1025 |
| 8. | Temp at D.S.O. | 655 | 680 |
| 9. | Assumed Drum Pressure | 1275 | 1275 |
| 10. | Sat Steam Temp | 578 | 578 |
| 11. | Btu/lb Shtd Steam | 1470 | 1457 |
| 12. | Btu/lb Dshtd Steam | 1278 | 1309 |
| 13. | Btu/lb Sat Steam | 1180 | 1180 |
| 14. | Btu/lb Feedwater | 249 | 249 |
| 15. | Btu/lb Feedwater to Sat Steam | 931 | 931 |
| 16. | Btu/lb Sat to Shtd Steam | 290 | 277 |
| 17. | Btu/lb Shtd to Dshtd Steam | 192 | 148 |
| 18. | Mm Btu/hr Feed to Sat Steam | 214.130 | 256.956 |
| 19. | Mm Btu/hr Sat to Shtd Steam | 66.700 | 76.452 |
| 20. | Mm Btu/hr Shtd to Dshtd Steam | -5.760 | -7.400 |
| 21. | Mm Btu/hr Total | 275.070 | 326.008 |
| | 15% x-s Air A/F Ratio = | 16.9 | 16.9 |
| 22. | Efficiency | 86.1 | 85.7 |
| 23. | Gross H V of Fuel NDFO | 19,300 | 19,300 |
| 24. | Lb Fuel/hr | 16,382 | 19,641 |
| 25. | Lb Air/hr | 276,856 | 331,933 |
| 26. | Lb Gas/hr | 293,238 | 351,574 |
| 27. | Furnace Volume | 1130 | |
| 28. | Btu Released/cu ft F V | 279,798 | 335,461 |
| 29. | Air Temp at Burners | 100 | 100 |
| 30. | Net H V of Fuel | 17,975 | 17,975 |
| 31. | Effective R H S | 422 | |
| 32. | Btu Available/sq ft R H S | 697,788 | 836,604 |
| 33. | Btu Not Absorbed/sq ft R H S | 523,000 | 639,000 |
| 34. | Btu Absorbed/sq ft R H S | 174,788 | 197,604 |
| 35. | Btu/lb Gas | 753 | 767 |
| 36. | Gas Temp lvg Furnace | 2752 | 2796 |

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CE 3938

FIRED MODE BOILER PERFORMANCE OF THE SCREEN BANK

April 5, 1972

| | | | |
|---|---|---|---|
| 1. | Tube Size & Thickness | 2 | O.D. x 0. 134 MWT In Line (✓) |
| 2. | No Rows Deep | 3 | Long Spacing 2 1/2 Stggd ( ) |
| 3. | No Tubes Wide | 32-31-32 | Trans Spacing 4 1/4 Prod. 10.62 |
| 4. | Rating % | 100 | 120 |
| 5. | Lb. Gas/hr W | 293,238 | 351,574 |
| 6. | Free Area | 75.8 | |
| 7. | Mass Flow | 3869 | 4638 |
| 8. | Eff Surface S | 595 | |
| 9. | Gas Temp ent $T_1$ | 2752 | 2796 |
| 10. | Est Gas Temp lvg | 2580 | 2636 |
| 11. | Avg Gas Temp | 2666 | 2716 |
| 12. | Cp | .318 | .319 |
| 13. | Sat Steam Temp Ts | 578 | 578 |
| 14. | Beam Length | 4.02 | |
| 15. | Rr | 3.08 | 3.12 |
| 16. | Diam Correction | .812 | .812 |
| 17. | Fa | .795 | .813 |
| 18. | Rc (Curve M-49B) | 15.27 | 17.3 |
| 19. | Rc Corrected | 9.85 | 11.42 |
| 20. | R | 12.93 | 14.54 |
| 21. | Aloge RS/WCp | 1.088 | 1.080 |
| 22. | $T_1$ - Ts | 2174 | 2218 |
| 23. | $T_2$ - Ts | 1998 | 2054 |
| 24. | Gas Temp lvg $T_2$ | 2576 | 2632 |
| 25. | Gas Temp ent Shtr | 2576 | 2632 |
| 26. | Btu/lb Gas ent Shtr | 697 | 714 |
| 27. | Mm Btu/hr Gas ent Shtr | 204.387 | 251.024 |
| 28. | Mm Btu/hr Absorbed as SM | 66.700 | 76.45 |
| 29. | Mm Btu/hr Gas lvg Shtr | 137.687 | 174.574 |
| 30. | Btu/lb Gas lvg Shtr | 470 | 497 |
| 31. | Gas Temp lvg Shtr | 1840 | 1931 |
| 32. | Draft Loss/Row | | |
| 33. | f | | |
| 34. | Draft Loss | | |

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Appendix D

## FIRED MODE BOILER PERFORMANCE OF THE SUPERHEATER

April 5, 1972

| | | | |
|---|---|---|---|
| 1. | Tube Size and Thickness | 1 1/2 | |
| 2. | No Tubes Wide and Spacing | 62@2 1/8 | |
| 3. | No Tubes Deep | 8 | |
| 4. | Effective Length/Tube | | |
| 5. | Effective Heating Surface S | 2210 | |
| 6. | Gas Free Area | 46.1 | |
| 7. | Rating % | 100 | 120 |
| 8. | Gas - Flow Wg | 293,238 | 351,574 |
| 9. | Temp ent $T_1$ | 2576 | 2632 |
| 10. | Temp lvg $T_2$ | 1840 | 1931 |
| 11. | Temp avg | 2208 | 2282 |
| 12. | Specific Heat Cp | .305 | .307 |
| 13. | Steam - Flow Ws | 230,000 | 276,000 |
| 14. | Temp ent $t_1$ | 578 | 578 |
| 15. | Temp lvg $t_2$ | 950 | 925 |
| 16. | Temp Rise | 372 | 347 |
| 17. | Enthalpy Change | 290 | 277 |
| 18. | Specific Heat Cs | .780 | .798 |
| 19. | Gas Mass Flow G | 6361 | 7626 |
| 20. | Transfer Rate R | 25.0 | 28.3 |
| 21. | RS / WgCs | .617 | .579 |
| 22. | WgCp / WsCs | .499 | .490 |
| 23. | $(T_1 - T_2) / (T_1 - t_1)$ | .410 | .394 |
| 24. | $T_1 - t_1$ | 1998 | 2054 |
| 25. | $T_1 - T_2$ | 819 | 809 |
| 26. | $T_2$ | 1757 | 1823 |
| 27. | $WgCp(T_1 - T_2) / Ws$ | 318 | 316 |
| 28. | Btu/lb Stm ent | 1180 | 1180 |
| 29. | Btu/lb Stm lvg | 1498 | 1496 |
| 30. | Shtd Stm Temp | 997 | 992 |
| 31. | No Steam Passes | | |
| 32. | Avg Specific Volume v | | |
| 33. | Actual Element Length | | |
| 34. | Equiv Length Bends, etc. | | |
| 35. | Total Equiv Length L | | |
| 36. | Lb Shtd Stm/min/Element F | | |
| 37. | Tube Inside Diameter D | | |
| 38. | $F^{1.85}$ | | |
| 39. | $D^{4.97}$ | | |
| 40. | Shtr. Pressure Drop | | |

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Appendix D

FIRED MODE BOILER PERFORMANCE OF THE MAIN BANK

April 5, 1972

| | | | |
|---|---|---|---|
| 1. | Tube Size & Thickness | 1 | Long Spacing 2 In Line (✓) |
| 2. | No Rows Deep | 28 | Trans Spacing 1 17/32 |
| 3. | No Tubes Wide | 91 | Product 3.0625 |
| 4. | Rating % | 100 | 120 |
| 5. | Lb Gas/hr W | 293,238 | 351,574 |
| 6. | Free Area | 48.7 | |
| 7. | Mass Flow | 6021 | 7219 |
| 8. | Eff Surface S | 7812 | |
| 9. | Gas Temp ent $T_1$ | 1840 | 1931 |
| 10. | Est Gas Temp lvg | 690 | 730 |
| 11. | Avg Gas Temp | 1265 | 1330 |
| 12. | Cp | .279 | .281 |
| 13. | Sat Steam Temp Ts | 578 | 578 |
| 14. | Beam Length | 2.42 | |
| 15. | Rr | 1.09 | 1.15 |
| 16. | Diam Correction | 1.0 | 1.0 |
| 17. | Fa | 1.013 | 1.011 |
| 18. | Rc (Curve M-49B) | 18.9 | 21.1 |
| 19. | Rc Corrected | 19.14 | 21.33 |
| 20. | R | 20.24 | 22.48 |
| 21. | Aloge RS/WCp | 6.91 | 5.92 |
| 22. | $T_1$ - Ts | 1262 | 1353 |
| 23. | $T_2$ - Ts | 183 | 229 |
| 24. | Gas Temp lvg $T_2$ | 761 | 807 |
| 25. | Gas Temp ent Shtr | | |
| 26. | Btu/lb Gas ent Shtr | | |
| 27. | Mm Btu/hr Gas ent Shtr | | |
| 28. | Mm Btu/hr Absorbed as SM | | |
| 29. | Mm Btu/hr Gas lvg Shtr | | |
| 30. | Btu/lb Gas lvg Shtr | | |
| 31. | Gas Temp lvg Shtr | | |
| 32. | Draft Loss/Row | | |
| 33. | f | | |
| 34. | Draft loss | | |

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Appendix D

FIRED MODE BOILER PERFORMANCE OF THE ECONOMIZER

April 5, 1972

| | | | |
|---|---|---|---|
| 1. | Tube Size and Thickness | 2" spiral | |
| 2. | No Rows High | 20 | |
| 3. | No Tubes Wide | 17 | |
| 4. | Length between Tube Sheets | 13.0 | |
| 5. | Heating Surface/Tube | | |
| 6. | Total Heating Surface S | 14,740 | |
| 7. | Rating % | 100 | 120 |
| 8. | Lb Gas/hr Wg | 293,238 | 351,574 |
| 9. | Lb Water/hr Ww | 230,000 | 276,000 |
| 10. | Gas Temp ent $T_1$ | 761 | 807 |
| 11. | Est Gas Temp lvg | 340 | 360 |
| 12. | Avg Gas Temp | 551 | 583 |
| 13. | Water Temp ent $t_1$ | 280 | 280 |
| 14. | Est Water Temp lvg | 450 | 460 |
| 15. | Avg Water Temp | 365 | 370 |
| 16. | Free Area (17)(13)(.134) | 29.6 | |
| 17. | Mass Flow | 9907 | 11,878 |
| 18. | Transfer Rate R | 11.6 | 12.8 |
| 19. | Cp | .258 | .258 |
| 20. | Cw | 1.054 | 1.057 |
| 21. | RS/Wg Cp | 2.260 | 2.080 |
| 22. | WgCp / Ww Cw | .312 | .311 |
| 23. | $(T_1 - T_2) / (T_1 - t_1)$ | .845 | .823 |
| 24. | $T_1 - t_1$ | 481 | 527 |
| 25. | $T_1 - T_2$ | 406 | 434 |
| 26. | $t_2 - t_1$ | 127 | 135 |
| 27. | Water Temp lvg Econ $t_2$ | 407 | 415 |
| 28. | Gas Temp lvg Econ $T_2$ | 355 | 373 |
| 29. | Avg Specific Volume | | |
| 30. | Lbs Water/Element/hr | | |
| 31. | Water Velocity | | |
| 32. | Length/Element | | |
| 33. | Econ Pressure Drop | | |
| 34. | Draft Loss/Ten Tubes | 5.0 | 7.4 |
| 35. | Total Draft Loss | 10.0 | 14.8 |

form 232 Calc. by Horlitz

Appendix D

## FIRED MODE BOILER HEAT BALANCE

April 5, 1972

| | | | |
|---|---|---|---|
| 1. | Rating | 100 | 120 |
| 2. | Ambient Air Temp Ta | 100 | 100 |
| 3. | Exit Gas Temp Tg | 355 | 373 |
| 4. | Air Temp to Burners | 100 | 100 |
| 5. | Lb Wet Gas/lb Oil | 17.9 | 17.9 |
| 6. | Lb Wet Air/lb Oil | 16.9 | 16.9 |
| 7. | Lb Moisture in Air/lb Oil | .279 | .279 |
| 8. | Lb $H_2O$ due to $H_2$ in Oil/lb Oil | 1.260 | 1.260 |
| 9. | Lb Dry Gas/lb Oil | 16.361 | 16.361 |
| 10. | H H V of Oil | 19,300 | 19,300 |
| 11. | Tg - Ta | 255 | 273 |
| 12. | (0.46)Tg | 163 | 172 |
| 13. | (1089) + (0.46)Tg-Ta | 1152 | 1161 |
| | Losses | | |
| 14. | Due to Moisture in Air | .173 | .185 |
| 15. | Due to Moisture from $H_2$ in Oil | 7.521 | 7.580 |
| 16. | Dry Gas | 5.187 | 5.553 |
| 17. | Total Calculated | 12.881 | 13.318 |
| 18. | Rad and Unaccounted for | 1.000 | 1.000 |
| 19. | Total | 13.881 | 14.318 |
| 20. | Efficiency by Difference | 86.119 | 85.682 |
| | | 86.1 | 85.7 |

CE 00107 37

Calc. by Horlitz

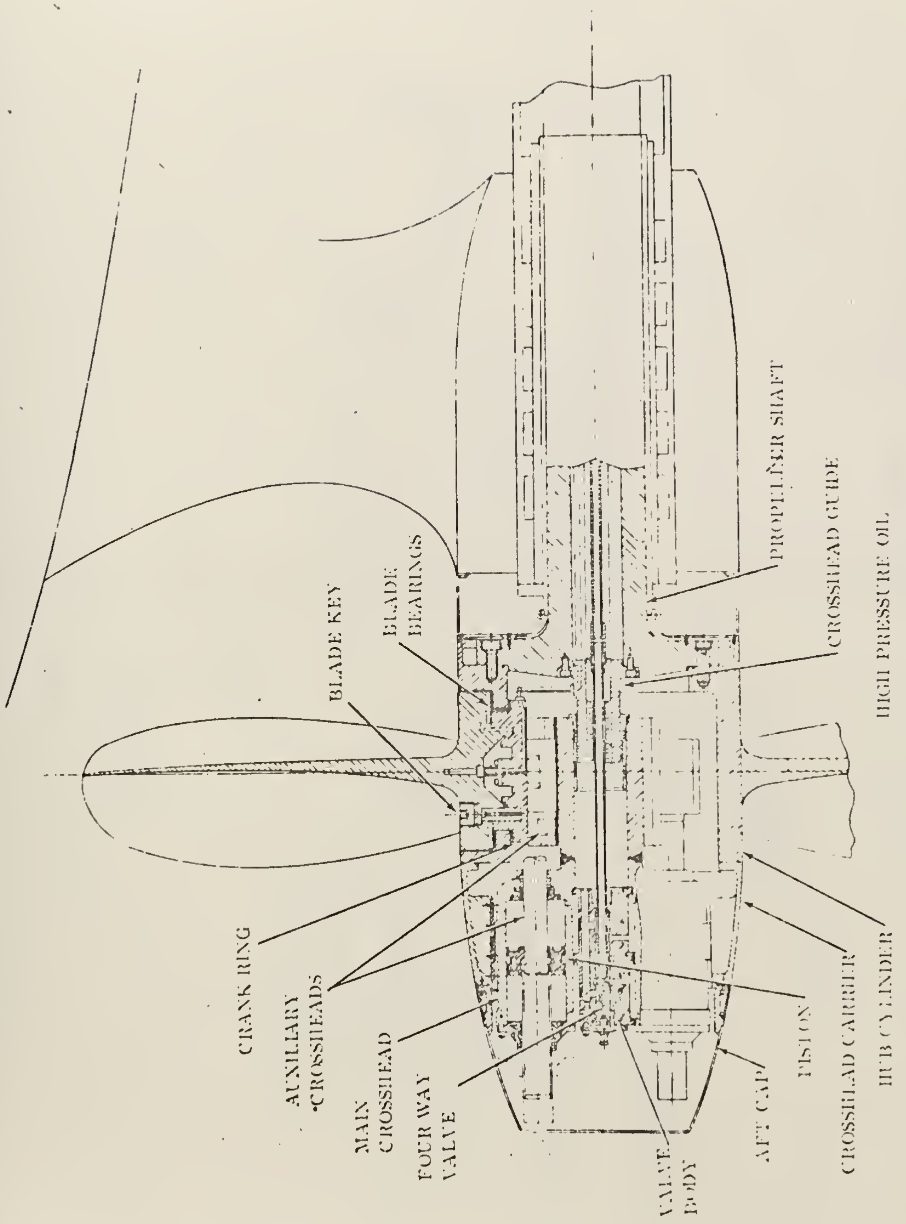



Figure 1. Hub and Blade Assembly

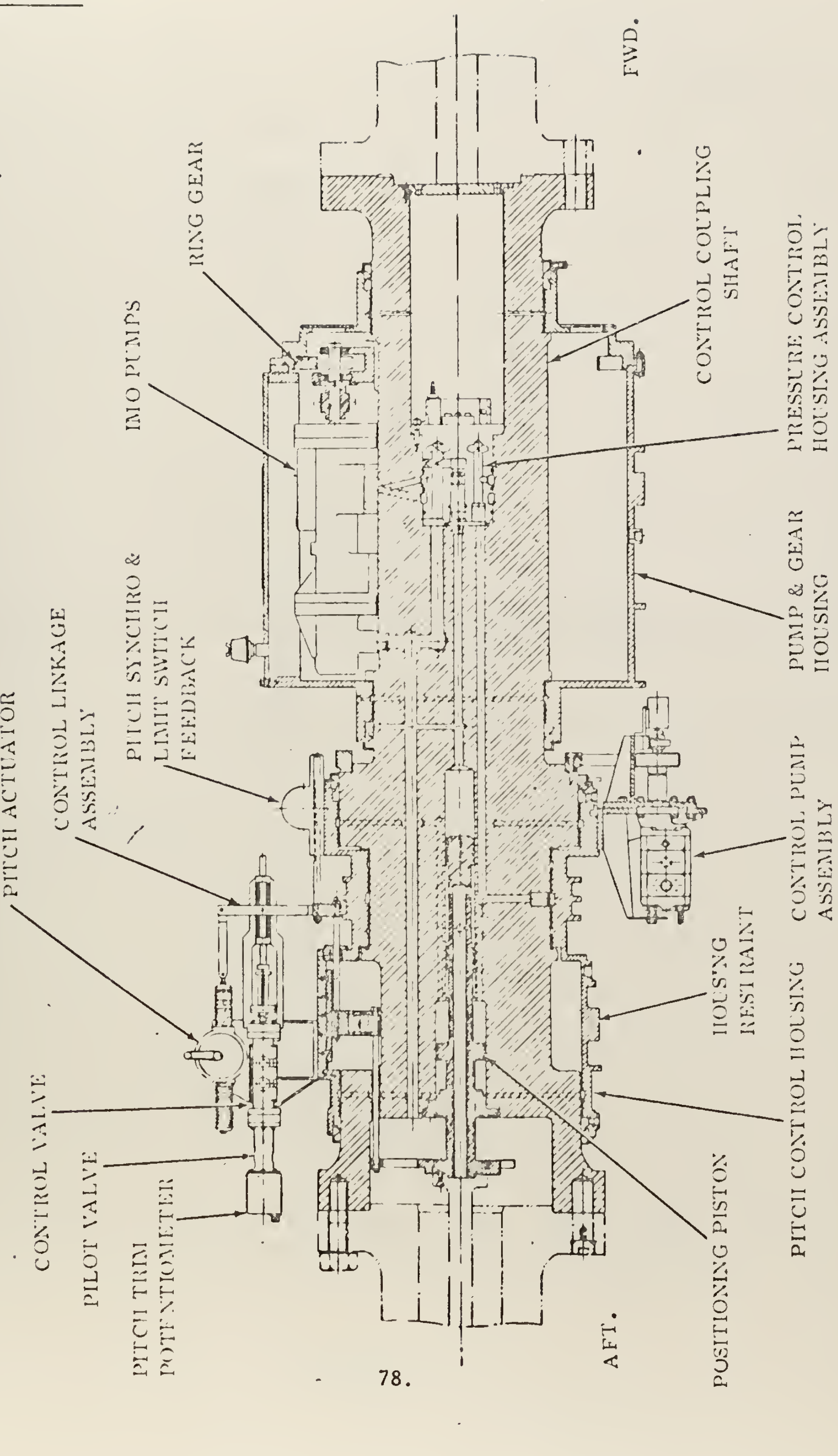
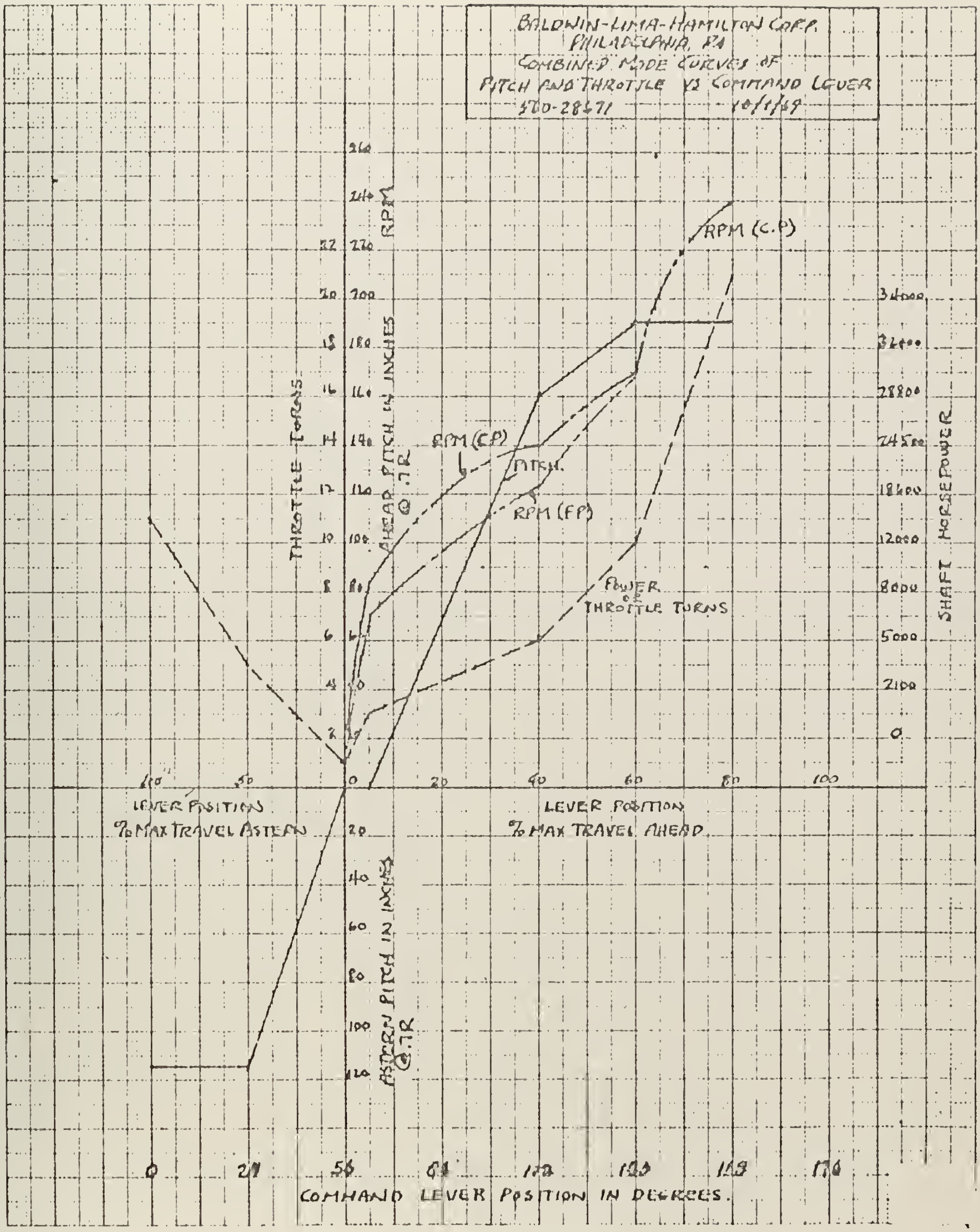
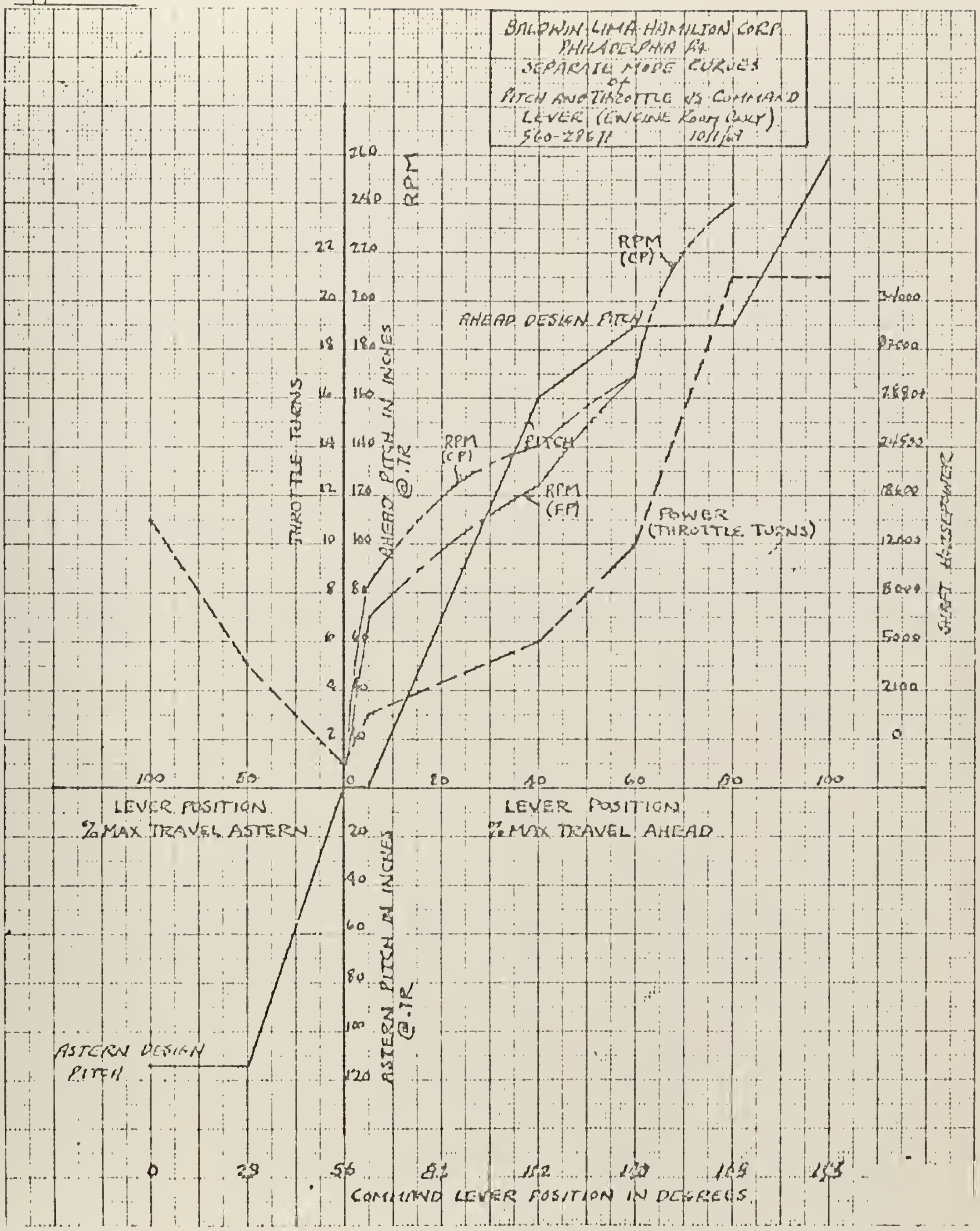



Figure 2. Control Coupling Assembly


BALDWIN-LIMA-HAMILTON CORP.
PHILADELPHIA, PA
COMBINED MODE CURVES OF
PITCH AND THROTTLE VS COMMAND LEVER
RPM (C.P)
RPM (CP)
PITCH
RPM (FP)
POWER
THROTTLE TURNS
THROTTLE TURNS
AHEAD PITCH IN INCHES @ .7R
RPM
SHAFT HORSEPOWER
LEVER POSITION
% MAX TRAVEL ASTERN
LEVER POSITION
% MAX TRAVEL AHEAD
ASTERN PITCH IN INCHES @ .7R
COMMAND LEVER POSITION IN DEGREES.

CURVE A


BALDWIN-LIMA-HAMILTON CORP.
PHILADELPHIA PA.
SEPARATE MODE CURVES
OF
PITCH AND THROTTLE VS COMMAND
LEVER (ENGINE ROOM ONLY)
RPM
THROTTLE TURNS
AHEAD PITCH IN INCHES @.7R
RPM (CP)
AHEAD DESIGN PITCH
PITCH
RPM (FP)
POWER (THROTTLE TURNS)
SHAFT HORSEPOWER
LEVER POSITION
% MAX TRAVEL ASTERN
LEVER POSITION
% MAX TRAVEL AHEAD
ASTERN PITCH IN INCHES @.7R
ASTERN DESIGN PITCH
COMMAND LEVER POSITION IN DEGREES

CURVE B

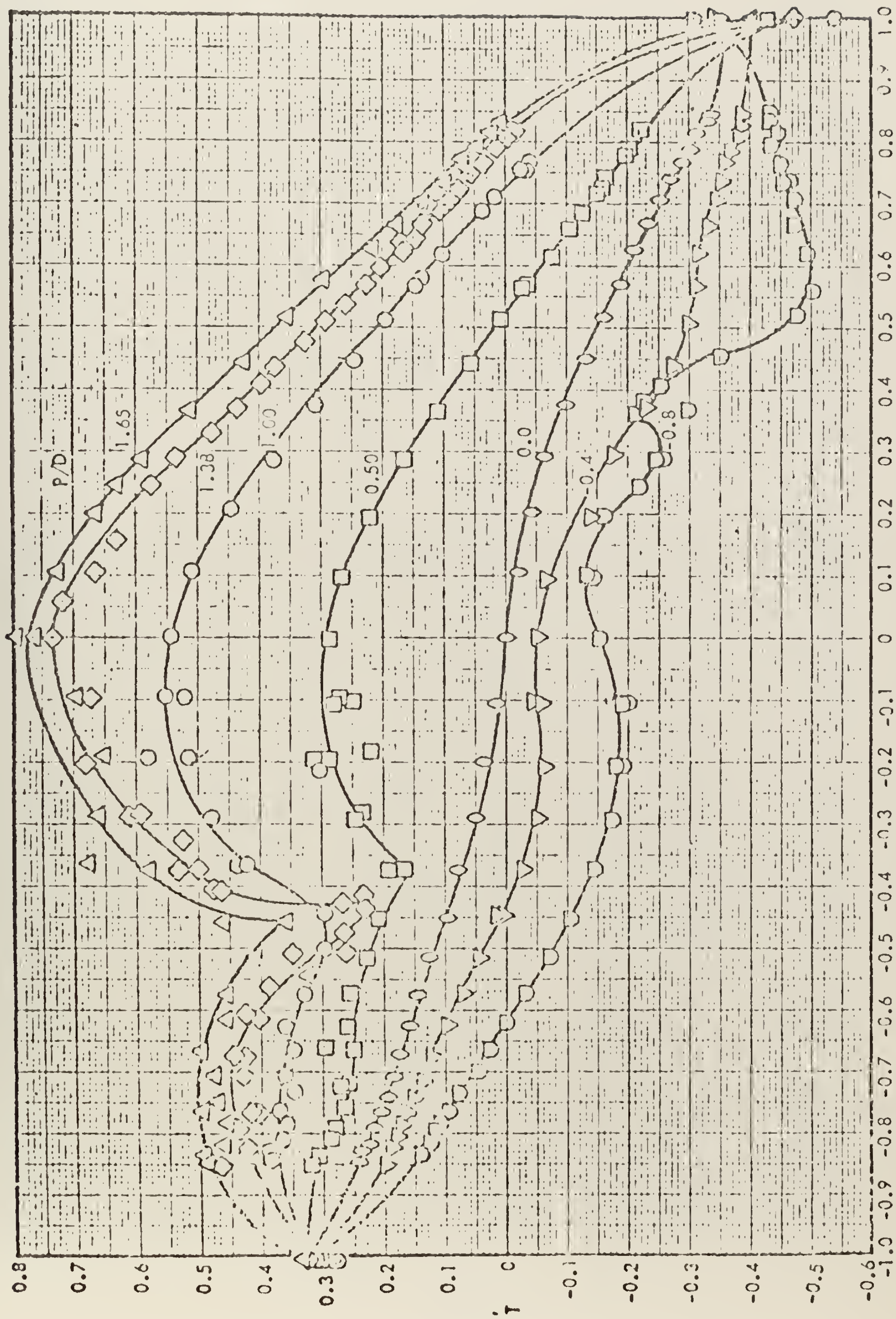



FIGURE 5 - CP PROPELLER THRUST CHARACTERISTICS.

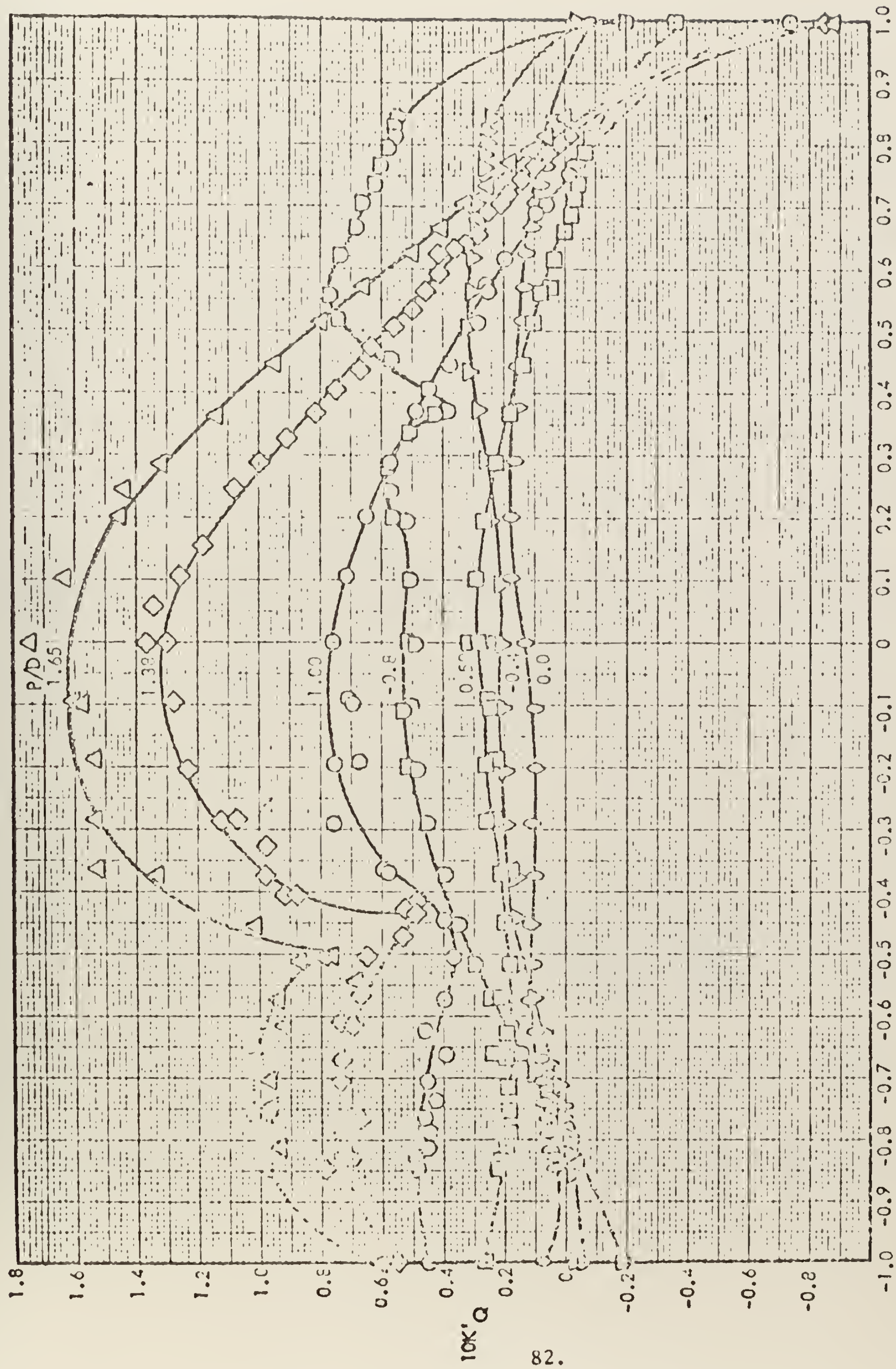



FIGURE 6 - CP PROPELLER TORQUE CHARACTERISTICS

Appendix E

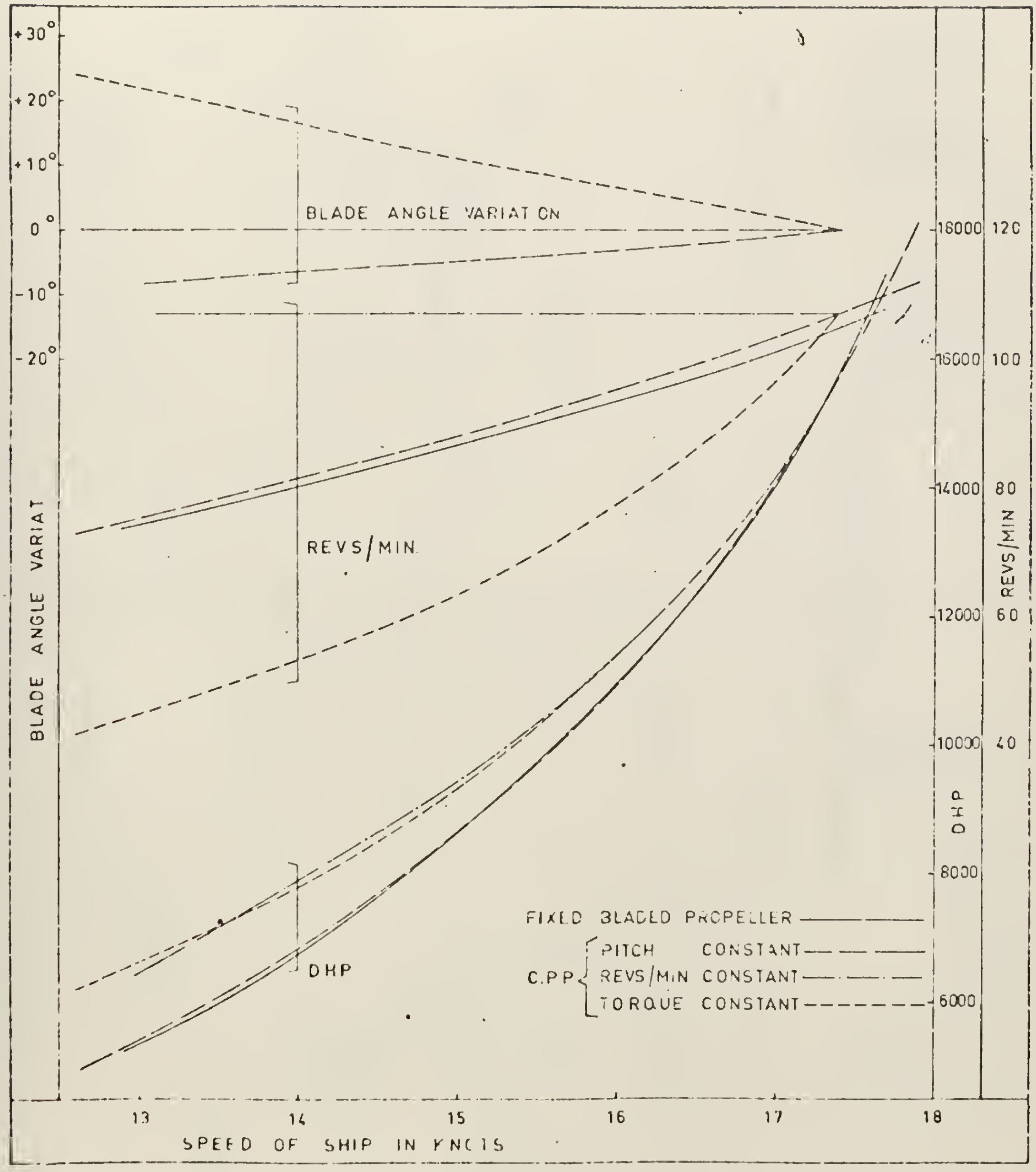
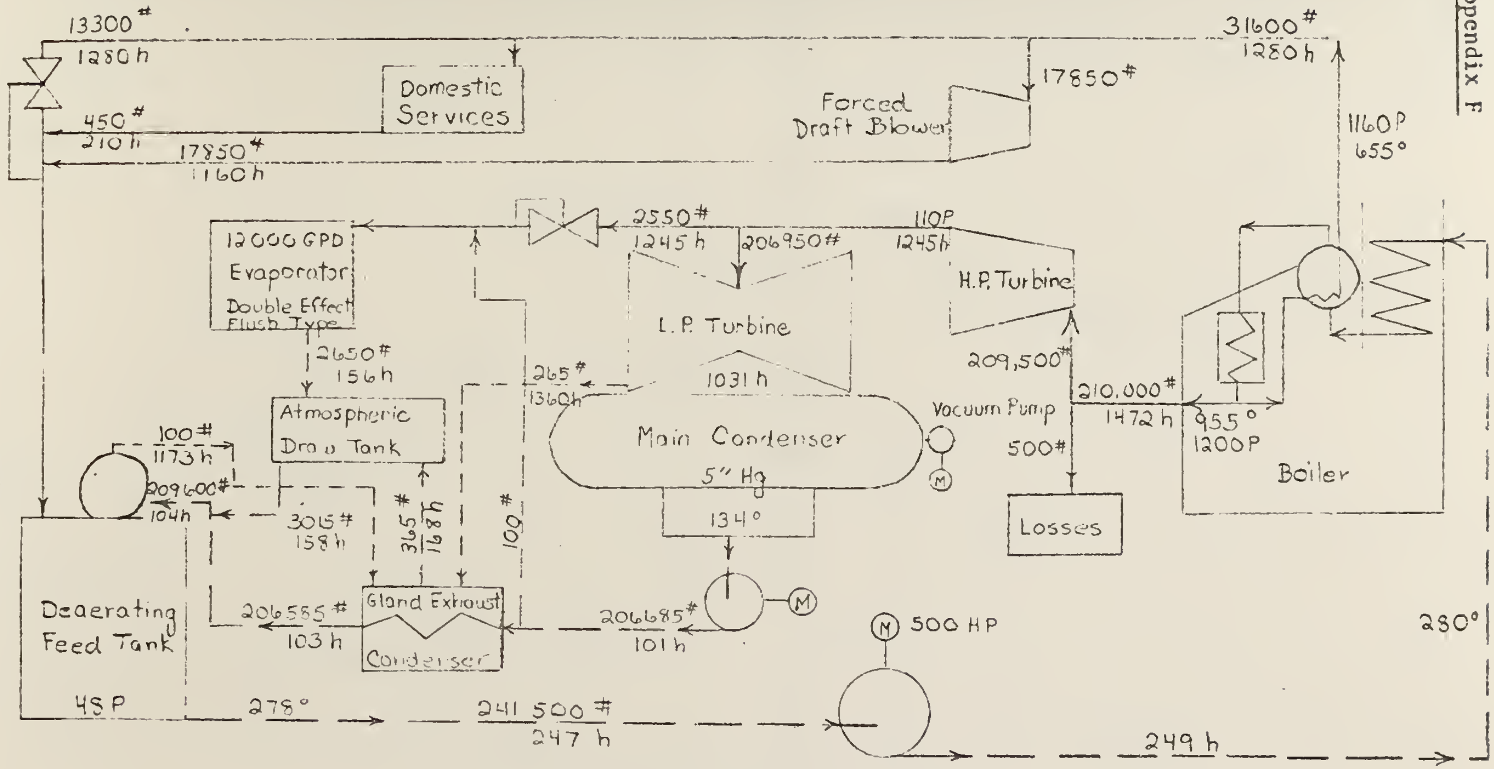



Fig. 8 Propulsion curves of Tanker when propelled by the fixed blade propeller and when propelled by the controllable pitch propeller: the latter respectively in the 0° blade setting (design setting), in settings for constant revs per min. and in settings for constant torque

Appendix F

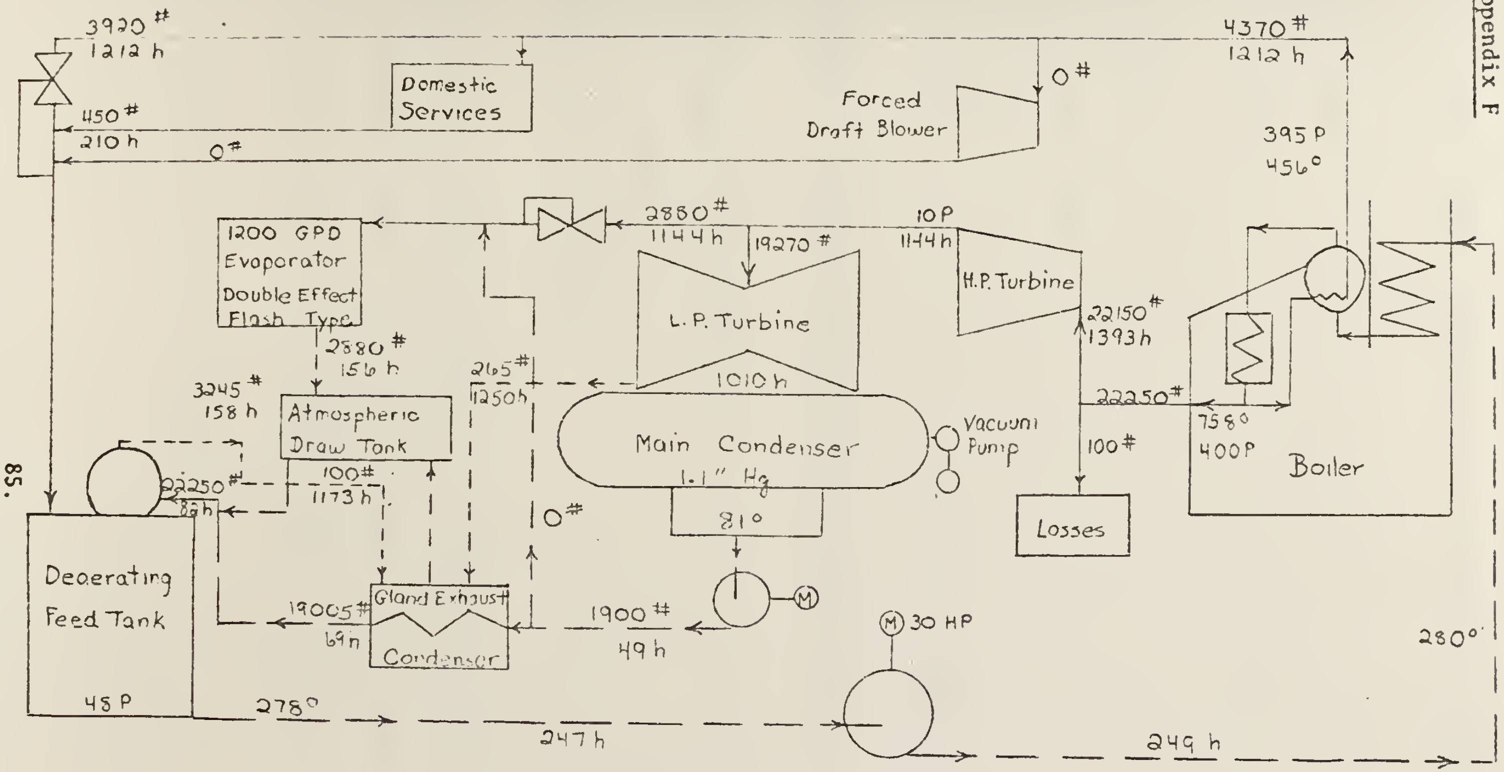



## Operating Conditions for Total Plant

| | | | | | | |
|---|---|---|---|---|---|---|
| Full Power: | HP Turbine | 18680 | Fuel (Distillate): | Boiler | 16382 | Lb/Hr |
| | LP Turbine | 17400 | | Gas Turbine | 3402 | Lb/Hr |
| | Gas Turbine | 5820 | | Elec.Gen.G.T. | 374 | Lb/Hr |
| | | 41900 | | (750KW/2) | 20158 | Lb/Hr |
| | After Gear Loss(600 HP) | 41300 | | s.f.c. | 0.484 | Lb/SHP-Hr |
| | After Attached Gen.Load (500 KW) | 41615 | | | | |

KW = 875 per plant

Appendix F




## Operating Conditions for Total Plant

| | | | | | |
|---|---|---|---|---|---|
| Cruise: | HP Turbine | 2165 HP | Fuel (Distillate): | Boiler | 0 Lb/Hr |
| | LP Turbine | 1015 HP | | Gas Turbine | 3402 Lb/Hr |
| | Gas Turbine | 5760 HP | | All Purpose SFC | 0.438 Lb/SHP-Hr |
| | | 8940 HP | | | |
| | After Gear Loss (500 HP) | 8440 HP | | KW = 500 per plant | |
| | After Taking Gen. Load (500 KW) | 7755 SHP | | | |

Appendix F

## CRUISE POWER CALCULATIONS
(based on Sheldon)

1. Percent exhaust heat theorectically available to Rankine Cycle

   A = 85.5 %

   B = 0.95      A x B = 81.2

   C = 0.85%      A x B x C = 69.0

2. Steam Flow

   F = 0.149 lb/lb

   G = 0.90

   E = 92%      $F \times G \times C \times \frac{E}{100} = 0.1053 \frac{\text{Lb steam}}{\text{Lb air}}$

   Steam Flow = Gas Flow x 0.1053 = 20,500 Lb/Hr

3. Step 1 corrected to 1.1 In. Hg

   D = 2 x 0.1053 = 0.211

   A x B x C - D = 68.8

4. Exhaust heat actually available to the Rankine Cycle at 1.1 In. Hg

   Available Heat (Ideal) = 225 BTU/Lb x 0.99 = 223 BTU/Lb

   Available Heat = $223 \times \frac{69.0}{100} \times \frac{92}{100} \times 194{,}000 = 27.1 \times 10^6$ BTU/Hr

5. Heat available to steam turbine

   J = 36.9 %

   K = 0.998

   Heat Avail. to Stm. Turbine = $(27.1\text{x}10^6) \times 0.369 \times 0.998 = 9.98\text{x}10^6$ BTU/Hr

6. Actual steam turbine output

Engine Efficiency = 0.79

$$\text{Output} = \frac{9.98 \times 10^6 \times 0.79}{2544} = 3100 \text{ HP}$$

11 SEP 77 24421
5 APR 79 25171

Thesis 135356
C5178 Clough
A cogas propulsion
cycle with peak effi-
ciency at low power.